INDEXED

Marine Engineering

APRIL, 1905.

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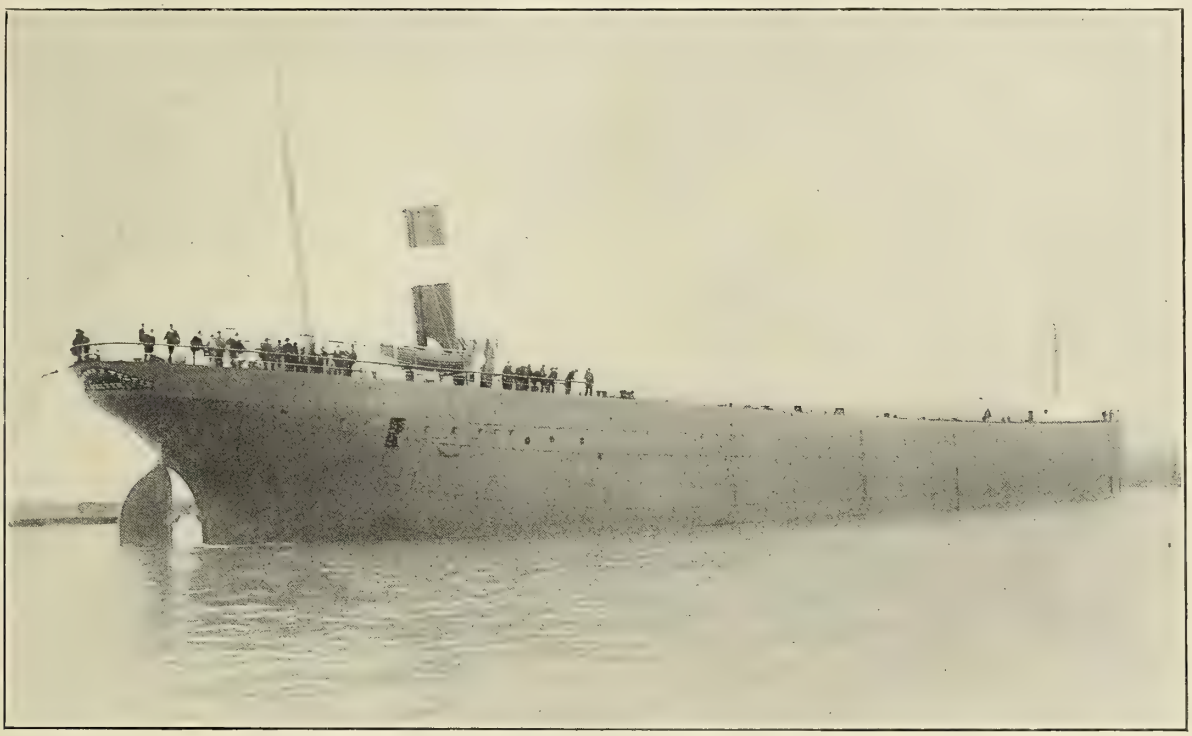
RECENT DEVELOPMENT OF LAKE SHIP BUILDING.

BY WALDON FAWCETT.

The placing of contracts for four new steel steamers, which will rank as the largest vessels ever constructed on the great lakes, emphasizes the present healthful condition of the steel ship building industry on the inland seas, and also furnishes further refutation of the contention that the fresh water cargo carrier has reached its maximum development of size under existent limitations in the depth of harbors and channels. These new freighters, and others for which contracts have recently been entered into, are of further interest by reason of the fact that they will constitute confirmation of the wisdom of certain rather radical innovations in design which were introduced by way of experiment in a preliminary vessel of this new class, turned out only a few months since.



THE LAKE STEAMER AUGUSTUS B. WOLVIN.

When, at the close of the last century, the management of the lake shipping interests, at that time controlled by John D. Rockefeller and his associates, entered into contract for the construction of several steel freight-carrying steamers of approximately 500 feet length over all, it was generally predicted that the acme of attainment had been reached, pending the deepening of certain waterways connecting the chain of lakes, the condition of which must ever, by reason of their positions, exert a determining influence upon lake navigation.

When the steamer *Morse* and other vessels of the Rockefeller fleet above mentioned were planned, it was claimed that the development which involved continual increase in the length and breadth of lake craft without a proportionate increase in the depth, had reached its limit. It was even predicted that progress in this direction must stop until there was available through the entire interior water highway a minimum depth of 23 feet of water, as compared with 18 or 19 feet, the average depth in the shallowest channels at that time. Moreover, so generally did this view of the situation appear to be accepted in shipping and shipbuilding circles that there was actually a revulsion of sentiment, and not only were no further contracts placed, for the time being, for vessels of the 500-foot class, but the policy of owners appeared to be to secure hulls ranging in length from 450 to 480 feet, and at this juncture not a few contracts were placed for vessels of from 436 to 450 feet.

This policy prevailed until the autumn of 1903, when the advance and development of the lake cargo carrier as a distinctive type of craft was resumed by the placing of a contract for the steamer *Augustus B. Wolvin,* a vessel 560 feet in length over all, and which ranks to-day as the largest ship in commission on fresh water anywhere on the globe. With the later increase in the size of lake vessels, however, has come a change in the methods of operation, which has in a measure made practicable the larger craft. In the days when the Rockefeller management reached the 500-foot limit in the length of ships, it was the policy of these interests to have each steamer tow one or two barges, in size nearly equal to the propeller, thus enabling one engine to move aggregate cargoes of about 20,000 tons of iron ore. Under the new *régime* this plan has been abandoned. The present-

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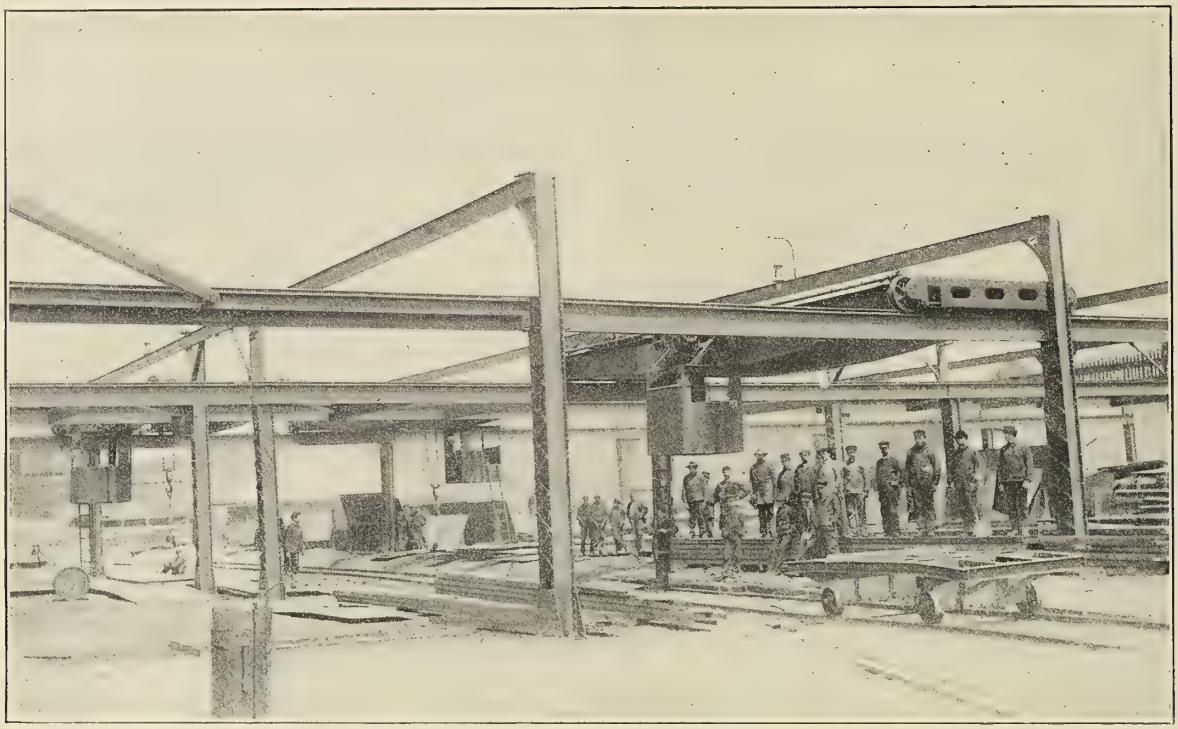
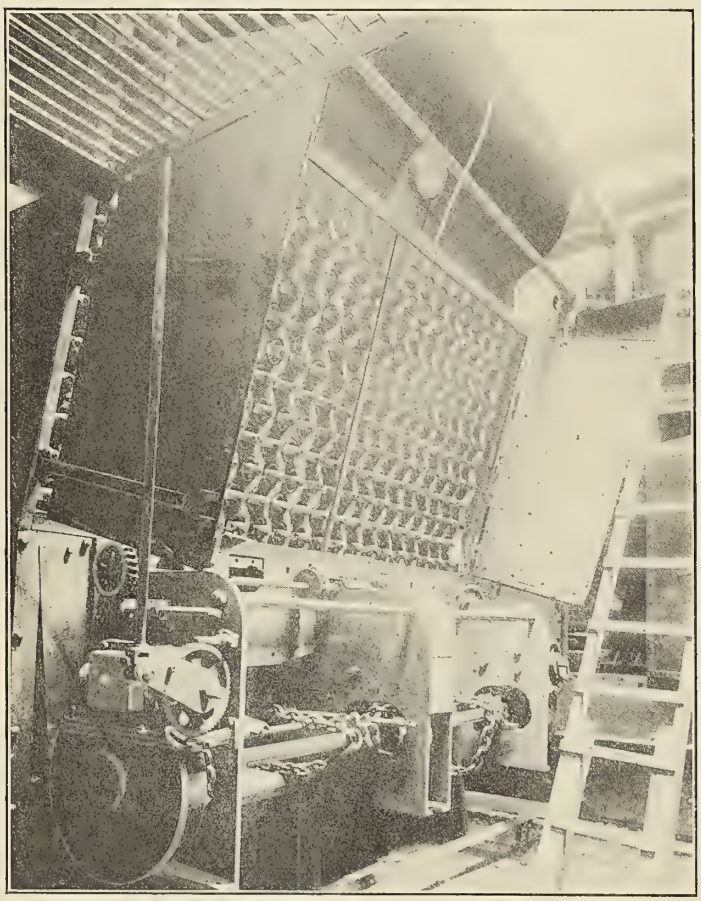


HANDLING MATERIAL IN A LAKE SHIPYARD.

day vessel of the greatest capacity is fitted with powerful engines, but it is not contemplated that she will attempt to handle tow barges. To compensate for the economical advantage found in the movement of three cargoes by one engine, the new design is for a steamer of 40 percent greater carrying capacity than any one of the old steamers or barges; capable of being operated at greater speed; with a vastly greater factor of safety in heavy weather; and with considerable saving for tug service and other handling charges, over that necessitated by the movement of vessels without power in rivers and harbors.



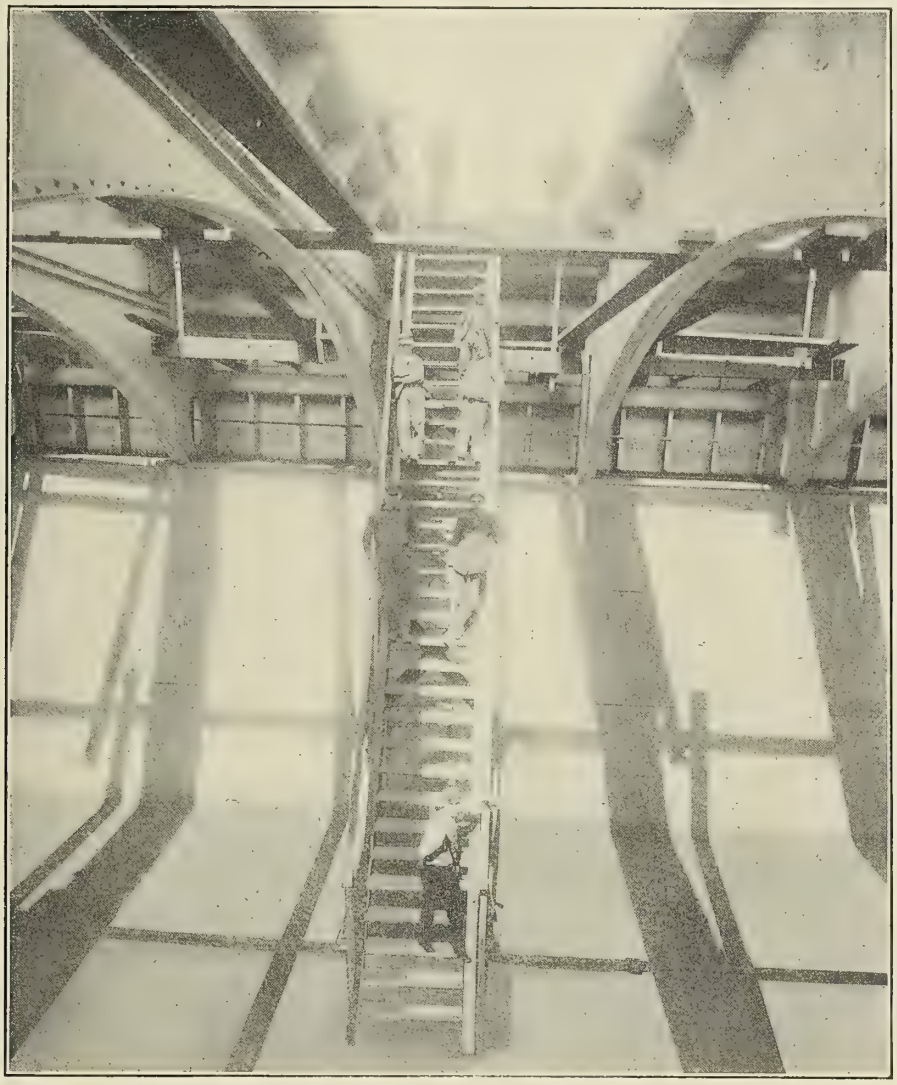
WATER-TUBE BOILER WITH AUTOMATIC STOKER.

The four new steamers are to be constructed for the Pittsburg Steamship Company, which constitutes the lake navigation department of the United States Steel Corporation. The new vessels will be virtual duplicates of the steamer *Wolvin,* save in length, in which they exceed the *Wolvin* by nine feet. Their dimensions will be 569 feet length over all, 549 feet length of keel, 56 feet beam, and 31 feet depth. The new vessels will cost $430,000 each, as compared with $480,000, the cost of the *Wolvin,* but it should be explained that this difference in cost is, in part, to be attributed to the provision on the *Wolvin* of elaborately-fitted owners' quarters.

The new class of lake steamers differ very radically from the older vessels in construction. This alteration of design has not resulted in any change in the always distinctive exterior appearance of the fresh-water freighters, but is plainly manifest in the arrangement of the interior. The new principle is known as the arch-hopper construction, and was inaugurated in the *Wolvin,* where it has proven so successful as to induce adoption for the four new steamers, and probably all future vessels of this class. This novel construction has not only contributed the additional strength essential in a vessel of the length of these, but permits the employment of the clam-shell or grab-buckets, which are rapidly coming into use for the automatic unloading at lake ports of the iron ore which constitutes the chief commodity carried by these vessels.

In the new form of construction the ordinary hold stanchions found in all the older types of lake vessels are entirely dispensed with, and in their place is substituted a system of girder arches which support the upper deck as well as the sides of the vessel. The cargo hold is built in the form of a hopper with sides that slope from the main deck down to the tank top and with ends built on the same slope. The extent of this incline may be appreciated from the fact that in the case of the *Wolvin* the width of the hopper at the top is 43 feet and at the bottom only 24 feet. Obviously such an arrangement is of immense advantage where it is desired to unload cargoes almost wholly by mechanical means. The arrangement of the cargo hopper in one continuous length without bulkheads or divisions of any kind, facilitates the

work of the automatic unloaders. Such an arrangement of cargo space with an open sweep of more than 400 feet is without parallel in the annals of steel ship construction. And, finally, the hatches of vessels of this design have been arranged with direct reference to the requirements of the automatic unloading machines. Each of the new vessels will have thirty-four hatchways (one more than the *Wolvin*) each hatch measuring 33 feet in length by 9 feet wide in the clear, spaced 12 feet centers, which space can be covered by the automatic ore unloaders working without assistance. Incidentally it may be noted that one of the most important of the subordinate improvements is found in the introduction of a system of steel plate sliding hatch covers, operated by steam power, resulting in a great saving of time and labor over the old plan of opening and closing the hatch covers by hand.



THE HOPPER BOTTOM OF THE WOLVIN, SHOWING MASSIVE GIRDER FRAMES.

The adoption of the arch-hopper type of construction has, of course, entailed a reduction of carrying capacity as compared with what would be afforded by a vessel of equal general dimensions, built according to the former design, but the advantages accruing from the new type more than compensate for any such sacrifice. Undoubtedly one of the most important features of the new type is found in the fact that it makes provision for the use of water ballast to an extent heretofore unheard of in lake freighters. Under the old plan of utilizing for cargo every foot of available space in a hull, which at best had not the depth which its length would suggest to be advisable, the only space for water ballast was in the double bottoms. With the lake type of freighter—a long vessel with all the engines and machinery at the extreme after end—the natural result was that boats running "light," or without cargo, were frequently kept in port, or at least had their progress greatly retarded in heavy weather. The new style cargo carrier, however, with its hopper hold affords, between the perpendicular sides of the hull and the inclined sides of the hopper, ample extra space for water ballast. Each of the four new steamers will have a capacity of 8,500 tons of water ballast, carried at the sides and in the double bottom.

The modern lake cargo carrier is never given very great engine power and the vessels of this new class have machinery of only 2,000 indicated horsepower, which is, however, sufficient to drive them fully loaded at a speed of 11 to 12 miles per hour. That there is considerable difference of opinion as to the best type of machinery to install is attested by the fact that the *Wolvin* has quadruple expansion engines with cylinders 18 1-2, 28 1-2, 43 1-2, and 66 inches in diameter by 42 inches stroke of piston, to which steam is supplied from two water-tube boilers of the Babcock and Wilcox type working at a pressure of 250 pounds per square inch, while each one of the four new steamers will be fitted with triple-expansion engines with cylinders 24 1-2, 39, and 65 inches in diameter by 42 inches stroke supplied with steam from two Scotch boilers working at 170 pounds pressure. However, a *résumé* of the large freighters now building or under contract in lake shipyards shows a preponderance of opinion in favor of the triple-expansion engines and the Scotch boiler provided with induced or forced draft.

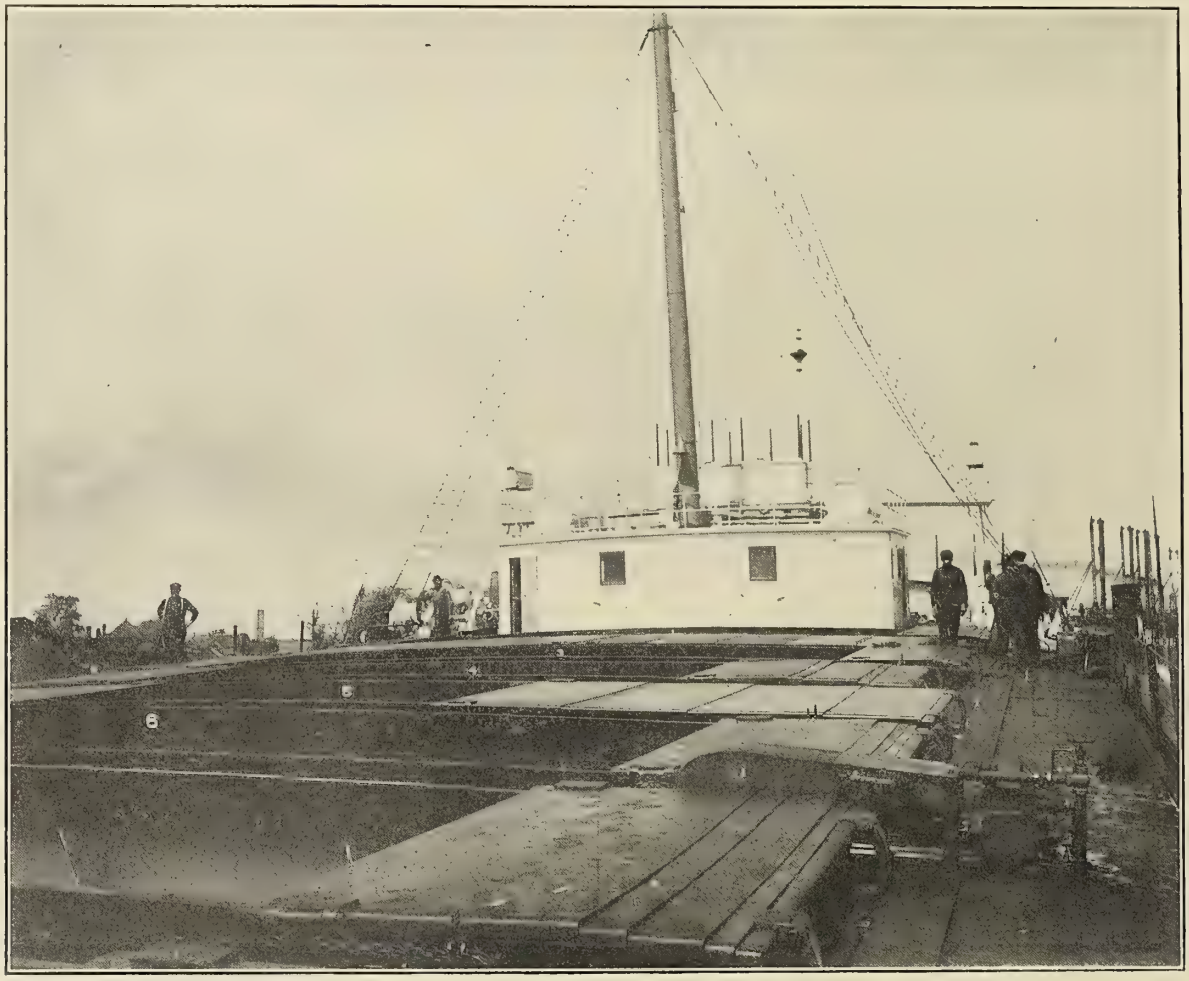
Each of the four steamers building for the United States Steel Corporation will have a carrying capacity of 10,000 gross tons on a mean draft of 19 feet, enabling the quartette of four vessels to move 800,000 tons of iron ore in a season of average length,

and increasing the aggregate carrying capacity of the entire Steel Corporation fleet to approximately 10,000,000 tons per season. That the new type of lake craft is neither an experiment nor the fad of one firm or individual, is attested by the fact that the opening months of the year 1905 see not less than seven of these 10,000-ton vessels under construction in lake yards for four different firms, while on the stocks at the same time are seven other cargo carriers ranging in length from 500 to 524 feet, and with carrying capacities of from 8,000 to 9,000 tons each. Indeed there is, at this writing, only one bulk freighter under contract on the lakes which is less than 500 feet in length or has a capacity below 8,000 gross tons.

While the new freighters of the 8,000 to 10,000-ton class are, almost without exception, to be constructed on the arch plan, the sides of the hopper holds will, in a few instances, be straight instead of sloping, thus curtailing, of course, the opportunities for carrying water ballast. The equipment of the new carriers shows an advance over former practice fully proportionate to the increase in the size of hulls. One notable innovation is the installation on each boat of two direct-connected electric plants of ample power to light every part of the ship. The illuminating plant is provided in duplicate in order to prevent inconvenience in the event of accident to either plant.

The recent placing of contracts for the new ships appears to mark the rejuvenation of lake shipbuilding, which has been in a somewhat depressed condition for more than a year past. The opening months of 1904 found under contract or construction in all lake yards vessels of an aggregate value of only $3,970,500. The indications are that the opening months of 1905 will find tonnage on the stocks or on the books of the ship builders to an amount fully double that of a corresponding date twelve months previous, and while the aggregate of the opening months of 1903 may scarcely be reached, the proportion of freighters in the whole building programme will probably be larger than in either 1903 or 1904.

The construction of several of these vessels constitutes an important factor in the iron and steel market, and, indeed, in the supply market generally. One of the new 10,000-ton cargo ships weighs 5,500 tons, and 750,000 rivets are used in her construction. In the construction of the four new freighters for the United States Steel Corporation there will be utilized about 14,000 tons of steel plates and small shapes, all of which will, of course, be furnished by the owners. Taking into consideration the needs of all the steel ship building plants on the lakes there is in sight a market for not less than 40,000 tons of shapes and plates.



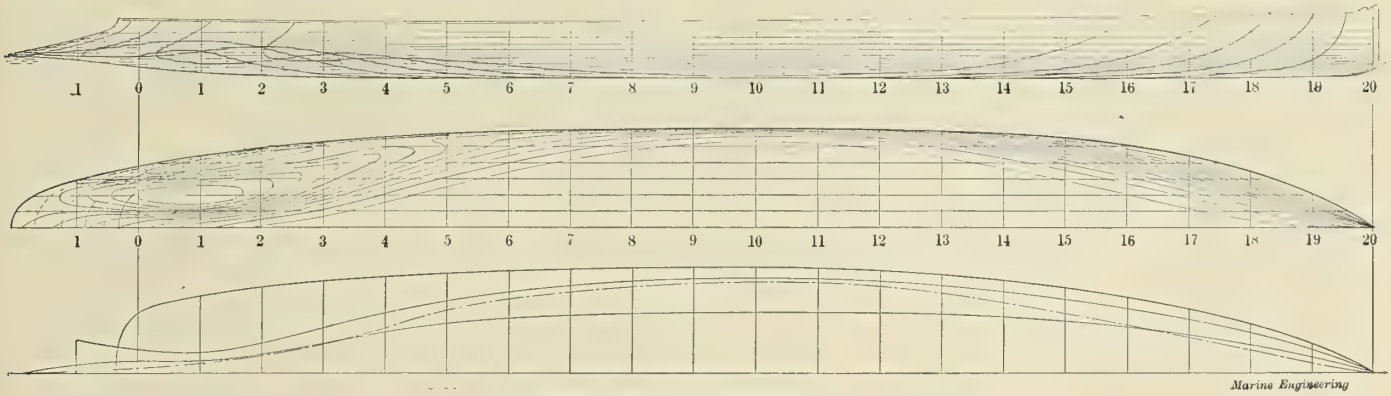
ON THE FORWARD DECK OF THE WOLVIN, SHOWING HATCHES.

The fourteen or more freighters of the new 8,000 to 10,000-ton class will be constructed for the most part by the American Ship Building Company, the consolidation controlling yards at Cleveland and Lorain, Ohio; Wyandotte, Mich.; West Bay City, Mich., and South Chicago, Ill., but contracts for several of the new type of craft are in the hands of the Great Lakes Engineering Works at Detroit, Mich. All the plants which will be engaged upon these contracts are essentially modern in equipment, qualified to handle this class of construction in the most expeditious manner possible, and it is anticipated that there may even be an improvement over the record made in the case of the pioneer 10,000-ton vessel, the *Wolvin*, which was launched in a total elapsed interval of 4 months and 9 days from the time the keel was laid, despite the loss of 20 days during the period by reason of floods and bad weather.

DESIGN OF A SHALLOW-DRAFT BOAT DRIVEN BY TWIN-TURBINE PROPELLERS.*

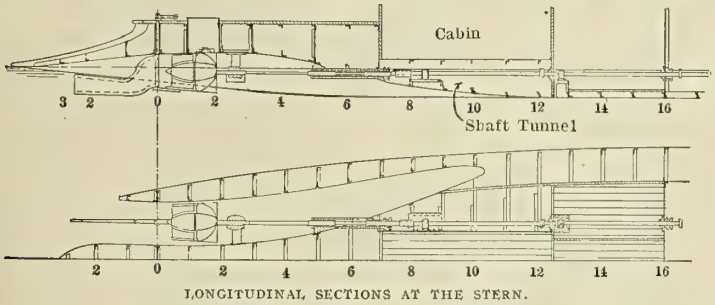
BY O. LIENAU.

The specification of this boat consists of the following problem: Required a fast boat of shallow draft for the purpose of inspection of a fortification located on a river. The draft of the vessel fully equipped, and provided with sufficient coal for 20 hours' steaming at full speed, shall not exceed 16 inches. The speed in slack water shall be 13 knots. The vessel shall have a cabin, with accommodations for five inspection officers, as well as other conveniences. A small mess room is also to be provided. It is optional to equip the boat with either a single screw or with twin screws, or to drive it with turbine screws. In the latter case it was stipulated that the turbine screws should be easily accessible.



LINES OF THE SHALLOW-DRAFT BOAT.

The choice of propellers determined the construction as well as the lines, and the entire installation. As it is almost impossible to obtain a speed of 13 knots with a single propeller on 15 3-4 inches draft, the twin-screw system was chosen, as with this system the feed of water to the wheels is much more favorable in shallow water than to a single screw located amidships. The turbine screw received the most favorable consideration, owing to the fact that an ordinary screw at 480 revolutions and 75 horsepower would have to be at least 24 inches diameter, of which about 10 inches would project above the water level. Under such a condition the screw would work very inefficiently, and even if it were placed in a carefully constructed propeller tunnel the feed of the water to the screw would be poor. A turbine propeller, however, will work well under such a condition, even if the center of the shaft is almost level with the surface of the water. The objection to the turbine propeller lies mainly in the fact that it is inefficient when going astern, but this objection is compensated for by the advantages quoted, and furthermore is reduced to a minimum owing to the fact that the directing vanes are only slightly inclined to the axis.



LONGITUDINAL SECTIONS AT THE STERN.

After deciding upon the style and design of the propellers, it was necessary to determine the dimensions of the vessels, of which there was made first an approximate, and later an exact, estimate of the weights, as follows:

1. Weight of hull, $L \times B \times H \times 231$ pounds, as a mean value, according to the experience of various firms.

2. Engine and Schultz boiler, 66 pounds per I. H. P.

3. Coal consumption at full speed, 2.1 pounds per I. H. P. per hour.

4. Equipment and crew, 1.5 tons, of which .75 ton represents equipment, and .75 ton corresponds to the weight of the crew of 10 men.

To determine the I. H. P. the French formula I. H. P. $= \frac{V^3 \varphi}{m^3}$ was used, and the factor was calculated to be equal to 2.65.

Dimensions of the vessel were calculated to be as follows:

Length between perpendiculars	69 feet 6 inches
Length over all	76 " 9 "
Greatest breadth molded	11 feet 0 inches
Draft of water	15 3-4 inches
Displacement	16.29 tons
Block coefficient	0.565
Midship section coefficient	0.92

$$\frac{L}{B} = 6.33, \qquad \frac{D}{B} = 0.12.$$

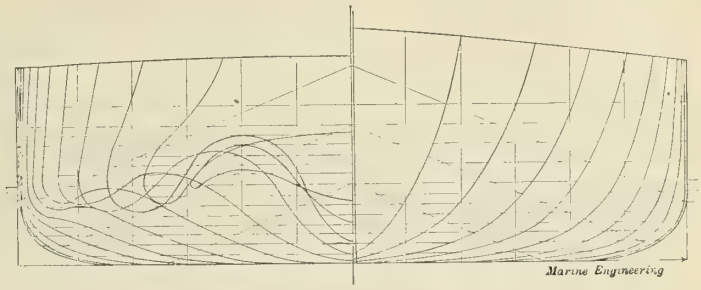
A speed of 13 knots is attained with two engines of a combined I. H. P. of 150, at 480 revolutions per minute.

Diameter of propellers, 23 5-8 inches.

Working pressure of Schultz boiler, 155 pounds per square inch, gauge.

Steaming radius at full speed, 20 hours.

Radius of action, 260 knots.



BODY PLAN OF THE BOAT.

Crew. 1 pilot, 1 engineer, 1 fireman, 1 boy, 5 officers, 1 orderly. Total, 10 men.

Total weights.

Hull	8.07 tons,	or 49.5	percent
Engines and boilers	4.36 "	" 26.7	"
Coal	2.88 "	" 17.8	"
Equipment and crew	0.98 "	" 6.0	"
Total	16.29 "	" 100.0	"

In the design of the vessel great attention was paid to the after body, as it was of the greatest importance to allow the water to flow freely to the turbine screws, and also to be properly discharged by these. In order to accomplish this it was decided to

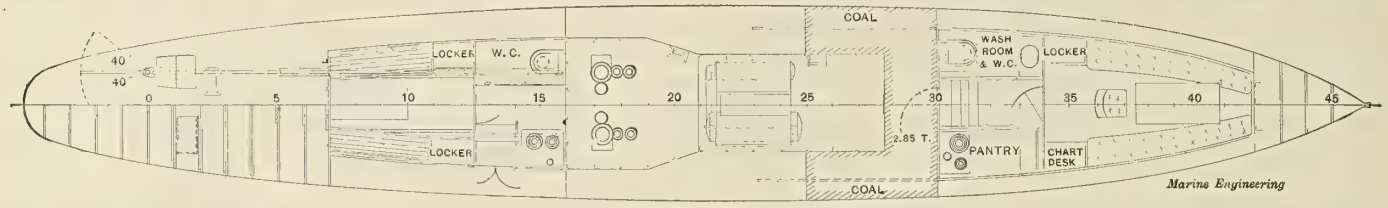
*Translated for Marine Engineering from *Schiffbau.*

permit the water to flow to the screws from the sides. The stern was given a long projecting shape similar to a duck's tail, which had the advantage that it gradually guided the water into the axial direction of the propellers. As the boat was equipped with two rudders, the keel could gradually grow smaller as it approached the stern, and the shape of the stern protected the rudders when landing. Furthermore, it was designed so that the center of gravity lay well aft, as the greatest weights were placed there. It was therefore located 4 inches forward of amidships. The lines of the forebody were of the customary shape, and arranged to give ample deck room for the cabin.

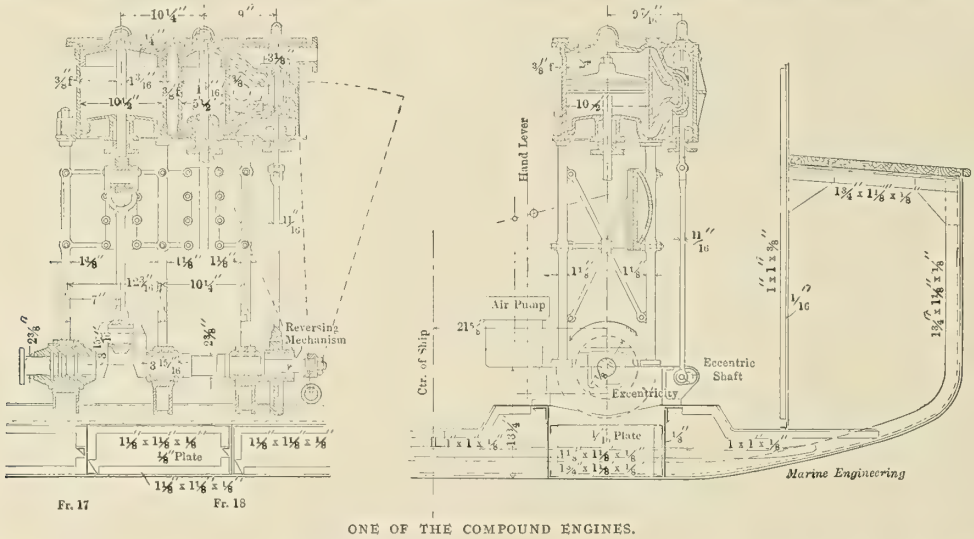
The distribution of weights had to be carefully considered. The officers' cabin was placed well forward, so that it balanced the weight of the machinery aft, and also afforded the inspecting officers an elevated position and an unobstructed view in all directions. The crew's cabin aft was built very light. The boiler was located at the fore-and-aft center of gravity. The forward cabin was fitted with a substantial table for five persons and sofas for accommodating seven persons, and, furthermore, with a large clothes locker and chart desk for fortification charts, and drawers under the sofas. The windows are particularly large in order to afford a broad outlook in bad weather. Aft of this cabin is located a toilet room with wash basin and water closet, and on the opposite side of this a pantry with a small stove and serving table. Sliding doors are fitted to the two last mentioned spaces. The entrance has a booby hatch and single door. On the port side a small ladder leads to the deck above the cabin, which space is fitted with two wooden seats and affords a view in all directions.

The fire room is surrounded by bunkers, the latter being all located beneath the deck, leaving ample room to fire. The engine space is quite roomy, allowing access to the engines from all sides. Both spaces are accessible by means of a single door and swinging hatch. The steering wheel is located above the boiler room, and a small pilot house, open at the back, protects the pilot from wind and weather. The boiler and engine spaces are separated from each other by a thin bulkhead.

The after cabin is fitted up for the crew of four men. The starboard door leads to a small galley, and a water closet is fitted on the port side, accessible by means of a sliding door. The cabin is fitted with a table, two settees, and two lockers. The engine shafts leading through this space run under the settees and are accessible by means of hinged covers. The two propellers can be withdrawn through two hatches in the deck, each hatch being fitted with light iron covers. Below the hatches are openings in the top of the propeller tunnels, which openings are fitted with watertight covers with rubber gaskets. In order to withdraw a propeller or a shaft, the shaft section under the settees is disconnected and the shaft drawn inboard, releasing the propeller. The guiding end of the shaft at the after end of the propeller can be drawn aft through the hub of the guide wheel.



ACCOMMODATION PLAN.



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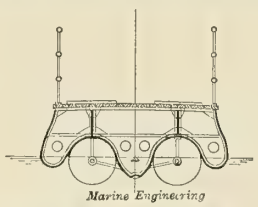
ONE OF THE COMPOUND ENGINES.



SECTION AT THE TRANSOM FRAME.

The material of the hull is the best ship's steel. In order to save weight as much as possible the scantlings were designed similarly to the requirements of the Germanic Lloyd, Class 1, 100 $\frac{A}{4}$ for inland vessels; although these sizes were reduced to some extent—not enough, however, to cause the stress to exceed 4,450 pounds per square inch in the outer fiber. The weakest place is the thwartship coal-bunker, which accordingly is braced by means of an angle-iron structure. The after-body had to be strengthened in various places, notably at the shaft bearings and propeller openings. The deck stringer has therefore been spread over the entire after-deck from frame 0 to frame 4, and braced at the middle. The floors were carried over the turbine tunnels. The rudders have a strong bearing, and the rudder heels are attached to the lower part of the guide wheels, which are very stiff, owing to their circular form.

The scantlings are as follows:

Length, 69 feet 6 inches. Breadth, 11 feet 0 inches.

Depth, 3 feet 3 3-8 inches. Draft, 1 foot 3 3-4 inches.

Stem, 3 inches by 25 pounds. Plate keel, 27 1-2 inches by 4 pounds.

Plating, 4 pounds. Sheerstrake, 19 3-4 inches by 4 pounds.

Deck stringer, aft, 19 5-8 inches by 4 pounds.
" " amidships, 37 3-8 inches by 5 pounds.
" " forward, 12 inches by 5 pounds.

Frames, 1 3-4 inches by 1 1-8 inches by 1-8 inch. Reverse frames, 1 1-8 inches by 1 1-8 inches by 1-8 inch.

Floors, at every frame, 3 1-2 inches by 4 pounds. Deck beams, at every frame, 1 3-4 inches by 1 1-8 inches by 1-8 inch.

Beam gussets, 4 pounds. Keel angles, 1 1-8 inches by 1 1-8 inches by 1-8 inch.

Frame spacing, 18 inches. Deck-house, 3 pounds. Deck-house angles, 1 inch by 1 inch by 1-8 inch.

Engine and boiler foundation, 5 pounds. Floor plates, 5 pounds.

Bulkheads, 3 pounds. Bulkhead angles, 1 inch by 1 inch by 1-8 inch.

Wood deck, 1 3-16 inches. Wooden walls of cabins, 3-4 inch.

Rivets, 5-16 inch and 3-8 inch diameter for plates 4 pounds and 5 pounds thick. Rivet spacing for outside plating, 3 1-2 diameters; and for frames, deck-beams, and seams not water-tight, 7 diameters. All is single riveting, with the exception of the cross-seams of the deck-stringers of 5 pounds thickness, where it is chain-riveted with butt-straps, spacing 4 diameters.

Calculations of the turbine screws.

The greatest diameter of the turbine screw which was permissible was 23 5-8 inches; revolutions, 480 per minute; horsepower of each engine, 75; and diameter of shaft, 2 3-8 inches. The following results are therefore obtained:

Area of circle	438.2	square inches
Sectional area of hub	4.4	" "
" " " blades	12.5	" "

This leaves a free sectional area of 421.3 square inches. The effective free sectional area was assumed to be 403 square inches.

Assuming the entrance slip to be 15 percent, the speed of entrance of water would be 25.82 feet per second, and the speed of the boat = 13 knots = 21.95 feet per second.

$$\text{Slip} = 15 \text{ percent} = \frac{w - v}{u} \times 100, \text{ and } w = \frac{v}{0.85} = 25.82 \text{ feet}$$

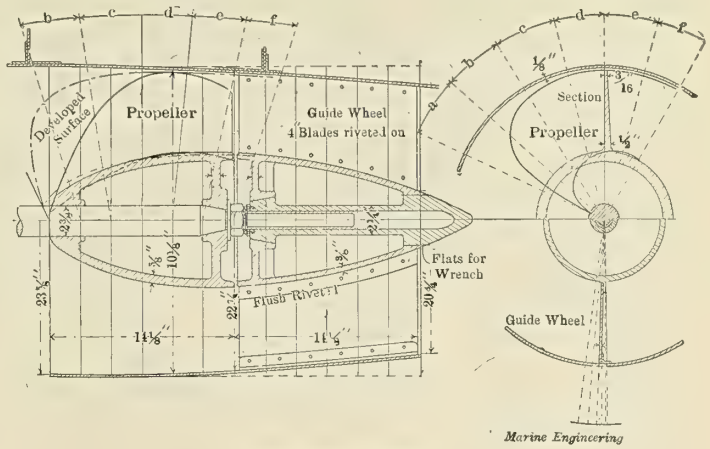
per second.

The speed of discharge was assumed, from previous experience, to have a slip = 45 percent. This therefore gives a speed of discharge of 39.37 feet per second. In the further design of the propeller it was determined that the speed of the water in the screw would increase according to the law of the parabola, and therefore the increasing pitch was designed according to the same law.

Furthermore, the cross-section of both the propeller and the guide wheel was also designed according to this law, as such a curve was well adapted to suit the shape of the tunnels. After the beginning and ending points of the parabolas were calculated they were graphically transferred to the drawings of tunnels. The hub sections were calculated by deducting from the total sectional area of the circle the sum of the free sectional area, the area of the cross-sections of the blades, and an allowance for friction and eddy formation.

The guide wheel was designed on the assumption that the rotation slip is 20 percent. The peripheral speed of the water at discharge is therefore:

$$\frac{\pi \times 25\text{-}8 \text{ feet} \times 480}{60} \times 20 \text{ percent} = 13.2 \text{ feet per second.}$$



THE TURBINE SCREWS.

The angle of discharge of the water as well as the angle of inlet to the guide wheel were then determined graphically and the latter direction was changed to an axial direction by means of a circular arc.

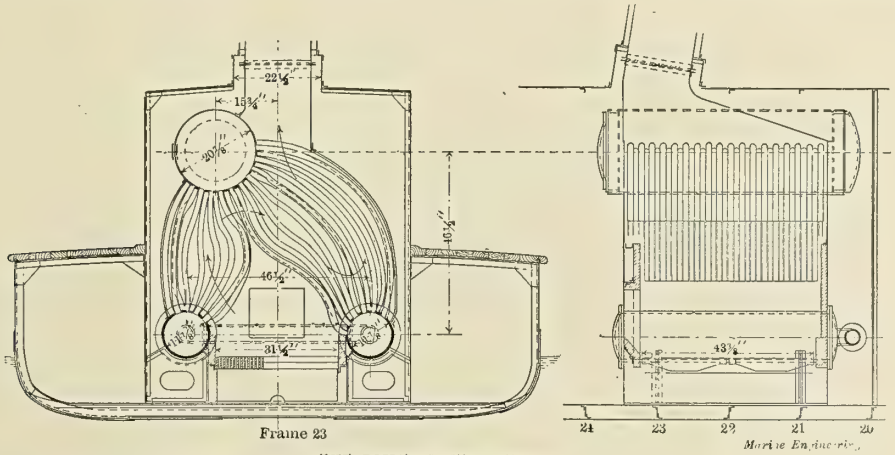
The propeller is made of bronze, the blades being cast onto the hub and those of the guide wheel being riveted on.

Schultz boiler.

Indicated horsepower per square foot of grate surface = 15.8.
Grate, 43 3-8 inches × 31 1-2 inches = 9 1-2 square feet.
Heating surface, 351 square feet.
Ratio, heating surface to grate area, = 37 to 1.

Boiler tubes, 176, of 1.181 inches outside diameter, 1 inch inside diameter; 192 of 1.417 inches outside diameter, 1.181 inches inside diameter; 368 tubes in all.

Water capacity, 1,100 pounds.
Steam pressure, 155 pounds per square inch.



THE BOILER INSTALLATION.

Area of stack = $\frac{1}{6.5}$ of grate area, = 1.46 square feet.

Two engines. Indicated horsepower, each 75.
High-pressure cylinder, 5 1-2 inches diameter.
Low-pressure cylinder, 10 1-2 inches diameter.
Stroke, 7 7-8 inches.
Pressure in receiver, 47 pounds per square inch.

Revolutions, 480 per minute.

High-pressure cut-off, down, 70 percent; up, 64 percent.

Low-pressure cut-off, down, 69 percent; up, 64 percent.

Angle of advance of eccentric, high-pressure, 40 degrees; low-pressure, 42 degrees.

Diameter of crank-shaft, 2 3-8 inches. Width of crank-slabs, 1 9-16 inches.

Diameter of crank-pin, 2 3-8 inches. Length, 3 1-8 inches.

Length of bearings, 3 15-16 inches.

Truss-block, 4 rings of 3 11-16 inches diameter, 1-2 inch thick.

Piston rod, base of thread, 13-16 inch; full diameter, 1 3-16 inches.

Connecting rod, diameter at top, 1 3-16 inches; at bottom, 1 7-16 inches.

Connecting rod, length = 5 cranks = 19 5-8 inches.

Piston thickness at hub, high-pressure = 1-4 inch; low-pressure = 3-8 inch.

Piston thickness at edge, high-pressure = 1-4 inch; low-pressure = 5-16 inch.

Three Ramsbottom Rings in piston 5-16 inch wide and 5-16 inch deep.

Cylinder walls, thickness 3-8 inch.

" flanges, depth 9-16 inch; width 3 1-8 inches.

Bronze cylinder covers, thickness 1-4 inch.

Number of studs in same, high-pressure 8; low-pressure 12.

The engines are compound with jet condenser. At the forward end of the high-pressure cylinder an eccentric is located which drives the high-pressure valve, and from this eccentric, by means of a combination of levers at the side, the low-pressure trick slide valve is also driven. Reversing the engine is effected by an arrangement located inside of the eccentric. The air pumps are driven by the low-pressure cross-heads, and are placed between the two engines in order to make them more accessible.

The stability calculations of the hull were made according to the method of McFarlane-Gray-Fellow. The stability quickly increases to 45 degrees, and then decreases quickly and reaches 0 at an angle of 67 degrees. This is caused by the fact that the side overhang of the boat acts as a pontoon and causes the stability to rise quickly at first, but as soon as these parts are submerged the increase is slight.

POWERING SHIPS.

BY EDWIN CERIO, NAVAL ARCHITECT.

PART IV.

VI. POWERING SHIPS BY MEANS OF MODEL EXPERIMENTS—EXPERIMENTAL TANKS.

As already said, if the total resistance of a ship's model is experimentally determined, by subtracting the part due to friction and multiplying the residual resistance by the cube of the ratio of similitude between ship and model, the *residual* resistance of the ship may be found; to this it is necessary to add the skin-resistance of the vessel in full size, thus obtaining the total resistance, from which it is easy to obtain the E. H. P., and, finally, the I. H. P. This is the method originally recurred to by Froude in the classical experiments carried out with the *Greyhound.** This vessel was towed by the *Active,* and the force developed by the traction was measured by a dynamometer at every run. Similar experiments were repeated on a model of the *Greyhound,* and the method described above was adopted to find the ship's resistance from that of the model. Having plotted out on a diagram the results obtained in both cases, the two curves representing the law of variation of resistance at different speeds practically coincided.

Model experiments are now carried out in experimental tanks. A laboratory in which very important investigations on ships' resistance have been made is that of Spezia, Italy. It was founded in 1887, and comprehends two distinct parts: that in which the paraffine models are prepared, and that in which they are tested. For the preparation of the models all the necessary contrivances are provided, such as the model-forming machine, a foundry, and a complete set of tools and machinery. The casting-box has the following dimensions: length 29 1-2 feet, breadth 4 1-3 feet, depth 2 feet 3 3-16 inches. The experimental tank, a transverse section of which is given in Fig. 10, is 480 feet in length. The North German Lloyd, after very satisfactory results obtained by placing models of its ships under experiment in Italy, decided to erect an experimental laboratory of its own; this was in 1899, and the experimental tank, which was the first in Germany, is in Bremerhaven. The laboratory covers an area of about 34,000 square feet, 22,000 of which are protected by sheds and roofs. A remarkable difference between this tank and nearly all others, is that it is constructed entirely of wood. Its dimensions are 540 feet in length, breadth 19 feet 8 1-4 inches, depth 12 feet 1 11-16 inches.

The preparation of the models is subject to special care; the paraffine, from which the castings are obtained, is melted in copper crucibles in which a tube is fixed, containing steam at a pressure of 10 pounds per square inch. As it is important not to heat the paraffine at too high a temperature, some experimenters prefer to melt it in a crucible surrounded by warm water (Bain-Marie); the best castings are obtained with paraffine heated at about 150 degrees Fahrenheit. The paraffine must also be purified before it reaches the casting-box; it is therefore made to riddle through a sieve of brass wire-gauze. The cast model is then put under the model-forming machine and shaved until it reproduces exactly, at a given scale, the form of the ship, and then it is ready for experiment.

TABLE XVIII.

Ratio of similitude between ship and model..ρ		Ship.	Model.
Corresponding speeds		V	v
Displacement		D	d
Length		L	l
Wetted surface		S	s
Resistance	due to skin friction	R_f	r_f
	" " wave configuration	R_w	r_w
	total	R	r
Constants		$C_t = \frac{R}{D}$	$C = \frac{r}{d}$
		$F_s = \frac{R_f}{D}$	$F_m = \frac{r_f}{d}$
		$C_w = \frac{R_w}{D}$	$C_w = \frac{r_w}{d}$
		$K = \frac{V}{D^{\frac{1}{6}}}$	$K = \frac{v}{d^{\frac{1}{6}}}$
		$M_c = \frac{L}{D^{\frac{1}{3}}}$	$M_c = \frac{l}{d^{\frac{1}{3}}}$
		$S_c = \frac{S}{D^{\frac{2}{3}}}$	$S_c = \frac{s}{d^{\frac{2}{3}}}$

MODEL EXPERIMENTS.

The method originally introduced by Froude is no longer used when testing models. The *method of constants* has taken its place, and will here be described.

Applying the law of dynamical similitude to the motion of ships it is possible to determine two functions, of resistance and speed, which depend only upon the speed of the ship and her shape, but are not influenced by her size. Thus, once the resistance of a ship's model is found by determining these constants, the results are applicable to any vessel which bears geometrical similarity to the model, provided the law of dynamical similitude is satisfied. To facilitate reference to the symbols employed hereafter,

*The *Greyhound* was an old corvette of the following characteristics: Length, 173 feet; breadth, 33 feet; draft, 13 1-2 feet; midship section area, 338 square feet; wetted surface, 7,540 square feet; displacement, 1,160 tons. Similar investigations to those of Froude had already been carried out in France, between the years 1840 and 1866, by Dupuy de Lôme and Bourgeois, with the ships *Sphinx, Pingouin, Duperré, Heroïne, etc.*

all the characteristics of ship and model, as well as the constants used, are gathered in Table XVIII; the meaning of these constants is explained in each case in which they occur.

Owing to the similarity of ship and model, and having assumed the speeds V and v as *corresponding speeds*, the following equations may be written:

$$\frac{D}{d} = \frac{R_w}{r_w} = \rho^3 \quad \ldots\ldots(1)$$

$$\frac{V}{v} = \sqrt{\rho} = \rho^{\frac{1}{2}} = \frac{D^{\frac{1}{6}}}{d^{\frac{1}{6}}} \quad \ldots\ldots(2)$$

therefore

$$\frac{R_w}{D} = \frac{r_w}{d} = \text{constant} = C_w \quad \ldots\ldots(3)$$

and

$$\frac{V}{D^{\frac{1}{6}}} = \frac{v}{d^{\frac{1}{6}}} = \text{constant} = K \quad \ldots\ldots(4)$$

But

$$R_w = R - R_f,$$
$$r_w = r - r_f,$$

so that the equation (3) may be written:

$$\frac{R - R_f}{D} = \frac{r - r_f}{d},$$

or

$$\frac{R}{D} = \frac{r}{d} - \frac{r_f}{d} + \frac{R_f}{D} \quad \ldots\ldots(5)$$

introducing the constants assumed:

$$C_t = C - F_m + F_s,$$

and

$$C - F_m = C_t - F_s = \frac{R}{D} - \frac{R_f}{D} = \frac{R_w}{D} = C_w \quad \ldots\ldots(6)$$

C and K are respectively the resistance and the speed constants, both independent of the ship's size; if a curve is plotted giving the values of K for every corresponding value of C_w, the resulting diagram expresses the law of variation of resistance in relation to speed for all vessels similar to the model. The constant, which applies only to that part of the resistance due to wave configuration, must be transformed into a constant C_t containing reference also to total resistance. The resistance due to friction may be expressed by the formulæ:

$$R_f = f_s S V^{1.825} \text{ and } r_f = f_m s v^{1.825}.$$

consequently

$$F_s = \frac{1}{D} f_s S V^{1.825} \quad \ldots\ldots(7)$$

and

$$F_m = \frac{1}{d} f_m s v^{1.825} \quad \ldots\ldots(8)$$

Having assumed $M_c = \frac{L}{D^{\frac{1}{3}}}$, the displacement $D = M_c^{-3}.L^3$ ..(9)

" " $M_c = \frac{l}{d^{\frac{1}{3}}}$, " " $d = M_c^{-3}.l^3$ ..(10)

" " $S_c = \frac{S}{D^{\frac{2}{3}}}$ the wetted surface $S = S_c.D^{\frac{2}{3}}$ ....(11)

" " $S_c = \frac{s}{d^{\frac{2}{3}}}$, " " " $s = S_c.d^{\frac{2}{3}}$ ....(12)

The relation between the corresponding speeds may be written:

$$V = v\sqrt{\rho} \text{ or, as } \sqrt{\rho} = \frac{D^{\frac{1}{6}}}{d^{\frac{1}{6}}},$$

$$V = \frac{v}{d^{\frac{1}{6}}}.D^{\frac{1}{6}} = KD^{\frac{1}{6}} \quad \ldots\ldots(13)$$

In the same way it may be found that:

$$v = Kd^{\frac{1}{6}} \quad \ldots\ldots(14)$$

Substituting the values given by the equations (9), (10), (11), (12), (13), and (14), in the equations (7) and (8):

$$F_s = f_s.D^{-1}.S_c.D^{\frac{2}{3}}.K^{1.825}.D^{\frac{1.825}{6}} = f_s.D^{-\frac{0.175}{6}}.S_c.K^{1.825},$$
$$= f_s.M_c^{\frac{0.175}{2}}.L^{-\frac{0.175}{2}}.S_c.K^{1.825} = f_s M_c^{0.0875}.L^{-0.0875}.S_c.K^{1.825}$$

the product $f_s.L^{-0.0875}$ may be considered as a coefficient depending upon the ship's length, and called O_s. Then:

$$F_s = O_s.M_c^{0.0875}.S_c.K^{1.825} \quad \ldots\ldots(15)$$

In the same way it may be found that:

$$F_m = O_m.M_c^{0.0875}.S_c.K^{1.825} \quad \ldots\ldots(16)$$

where $O_m = f_m.L^{-0.0875}$

By means of these equations it is possible to express the total resistance, the wave- and skin-resistance, with known quantities and experimental factors as follows:

$$R_f = F_s.D.$$
$$R_w = C_w.D = (C - F_m)\,D. \quad \text{(See equation 6.)}$$
$$R = C_t.D = (C_w + F_s)\,D. \quad (\text{" " 6.})$$

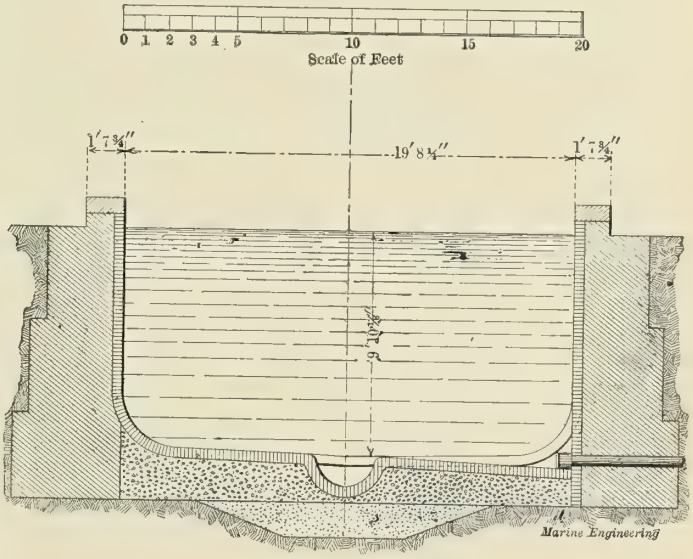


FIG. 10.

Therefore, when the resistance of a ship has to be determined with model experiments, the part due to wave configuration is obtained very simply by multiplying the quantity C_w by the displacement; that due to friction is found first by reading the value which has to be assumed for the product $f_s.L^{-0.0875} = O_s$ in a table where such values are calculated for all lengths of ships; then by multiplying this coefficient by the quantity $M_c^{0.0875}.S_c.K^{1.825}$, and by the displacement.

It is understood that the speeds must satisfy the condition established by equation (13). This method proves very practical whenever a tank can be disposed of; should the model be tested in fresh water the necessary correction must be made, and care must be given to the choice of appropriate units to measure the different elements which enter in the formulæ given.

Making Large Wrenches in Emergencies.—For turning nuts on large unions where there is no wrench of a size suitable for the work in hand, it is not always necessary to forge a wrench. For a brass union on a 2 1-2-inch pipe, take two pieces of iron about 1-2 inch by 2 1-2 inches and 18 inches long and two 1-2-inch bolts.

Bore holes for the bolts and screw up as shown at Fig. 1. The two projecting ends coming on opposite sides will serve as handles for turning. Friction is reduced by applying force on two sides of the pipe. Where there is not room enough to use this wrench, heat a single piece of iron, bend it into the shape of Fig. 2 and join with one bolt. Turn by means of the long handle.—*The Fulcrum.*

MAINTENANCE OF MACHINERY IN MERCHANT SHIPS.*

BY ROBERT HAIG.

The maintenance of machinery in merchant ships is a problem that is steadily becoming of higher importance by reason of the increase in size and power, the greater complexity and detail now found on board high-class vessels, and also because of the greater demands now made for a higher efficiency in the machinery, and the personnel of the engineering staffs afloat and ashore. The present-day ships call for engineers of wide experience, men of good education, who have had a thorough practical training for their business, combined with sufficient theory to enable them to appreciate heat losses, combustion, feed heating, electrical and refrigerating installations, and the many other duties that go to make up the experienced engineer, in whose hands in conjunction with the shore engineer superintendent the maintenance of valuable shipping property is entrusted. In dealing with this subject there will also be considered the various changes, improvements and developments that experience gained in maintenance has brought about.

Engine Foundations.—The practice now usually followed where a vessel is fitted with a double bottom is to put the engines right down on the tank-top, the scantlings of the inner bottom having been made heavier for that purpose. Formerly it was the practice to build a seating of deep girders and intercostals securely riveted and bracketed to the tank-top plating and the framing underneath. This made a very satisfactory foundation when well built, and it had the further advantage that the heads of the holding-down bolts could always be got at. The ruling practice at present, however, is to put the engines down on the tank-top plating, and it is here very serious trouble is met with by holding-down bolts breaking and becoming loose, by engines working on the seatings and causing serious trouble, and this will always ensue unless the highest quality of workmanship is put into the double bottom in the machinery space.

It is not suggested that placing the engines down on the tank-top is an inferior arrangement to the built-up seating; it is inferior only when adequate provision has not been made for strengthening the tank-top and double bottom at this part.

As is well known the scantlings of the double bottom in the machinery spaces are considerably increased, double reverse frame bars are fitted, and altogether the double bottom at this part is, or should be, of superior construction to what is required at other parts of the vessel. It is but rarely, however, that the material fails; the usual trouble is from bad workmanship, such as unfair holes and bad riveting produce, and that in a place where nothing but the best possible workmanship will stand. The intercostals should be well fitted; the rivet holes should be absolutely fair; the countersinking of the holes in the tank-top plating should be almost, if not the whole, depth of the plating, and the rivets laid up with good full points, so as to ensure a thoroughly strong connection. When we consider what a pull must come on this work when the engines are swinging about, it will be seen how very necessary it is that the holding-down bolts should take hold of a good foundation.

Care should be taken in spacing the holding-down bolts that they go straight down, not at an angle (Fig. 1); also that they do not come down on the heel of the frame bars, as in that case they have to be tapped into the top plating without being reinforced with a nut underneath, and it is these unfair bolts that are most frequently broken in working; and, unfortunately, broken and loose holding-down bolts are altogether too common. As so much of the successful working of the engines depends upon the design and workmanship of the engine seating and fastenings, the vital importance of this point cannot be too highly rated, nor the demand for good workmanship too rigidly enforced.

Shafting.—Although last winter was one of the most severe of recent years, the number of broken crank shafts was comparatively few when compared with the failures of past years, and this is not surprising and can readily be explained by the higher quality of material of which shafts are now made, by the increased attention given to the bearings and keeping the shafts in alignment, and also, no doubt, to the improved design and construction of the engine bed-plate foundations.

Although for very sufficient reasons solid crank shafts are still used in war ships, they are in merchant ships a thing of the past, and we have instead built crank shafts of nickel and carbon steels of the highest quality, well designed and constructed, with the result that broken crank shafts are exceedingly rare.

In the early days of built shafts no doubt there were many failures, and loose crank pins and unfair shafts were common, but higher quality of material, greater experience and accuracy in machining, improved methods and knowledge of shrinking the various members together, have resulted in producing shafts as nearly mechanically perfect as could be desired.

Built shafts are heavier, but that is of little importance in merchant ships, when compared with the higher quality of shaft, lower cost of manufacture, and also the very much reduced time required to renew.

Propeller shaft troubles have been numerous and their maintenance comes very high, by reason of the delay caused by repairs, cost of docking, etc. If we had only the torsional stresses set up by the power transmitted from the engines to deal with, this could be readily met by increasing the size of the shaft until the

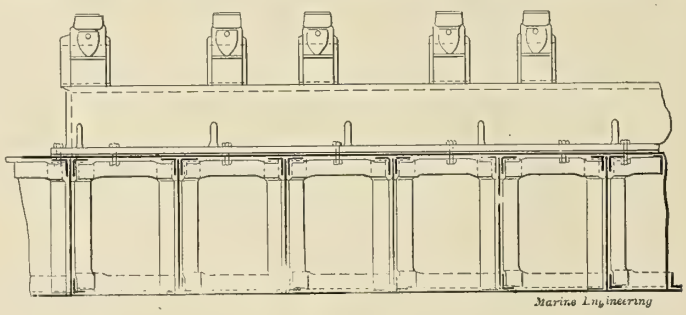


FIG. 1.

weakness was overcome; but a brief analysis will show that the stresses borne by the propeller shafts are of a widely varying nature. We have the stresses set up by the overhung weight of the propeller, which in some cases reaches 25 tons; also the bending stresses due to the change of the center of effort brought about by the continuous variation in the immersion of the propeller, and which reaches its maximum when a vessel is pitching heavily or in a more or less light condition; and, further, the abnormal stresses developed by the rapid acceleration and as sudden retardation of the momentum of the propeller when the engines are racing, all combine to produce conditions that demand a very high degree of efficiency both in the material and workmanship of the propeller shaft and stern bearings. There is the further troublesome factor to be considered, namely, the wastage on the propeller shaft at the ends of the brass sleeves due to the so-called galvanic action or fatigue; but happily this latter feature of the trouble has been to a large extent eliminated of late by protecting the shaft with brass sleeves for its entire length, thus doing away with what has hitherto been a serious menace to the life of propeller shafts.

In some cases the brass sleeves are put on in one length, which course is much to be desired, but where the shafts are long and the sleeves require to be built up of several lengths the greatest care should be observed to have the joints of the sleeves perfectly water-tight, as leakage at these parts causes scoring, and the shaft may rapidly become unsound without giving any outward evidence of the fact.

Considerable success has been attained by running propeller shafts without any brass sleeves whatever; the stern tube being

*Paper read at the twelfth annual meeting of the Society of Naval Architects and Marine Engineers, New York, November, 1904.

sealed at the outer end by a patented arrangement, which allows for the stern tube being filled with oil.

White metal is fitted into the stern bush instead of the ordinary lignum-vitæ, and the stern tube being filled with oil this makes an oil bath for the shafts and bearing and reduces wear and tear to a minimum.

Under these conditions the S. S. *Clematis* was in commission for five years and nine months, in which time the vessel steamed a distance of 156,000 miles, and when the shaft was drawn for examination the wear on the stern bearing was found not to exceed 1-16 of an inch and on the shaft it was practically *nil,* with the surface of the shaft preserved clean and bright and free from any wastage.

The arrangement of running shafts without any brass sleeves also allows for a shaft of larger diameter being used without increasing the size of the stern tube.

Greater attention is now given to the stern bearings of ships by increasing the length of the bearing, and also by keeping the shafts better in line.

In some long ships of high power flexible couplings have been fitted in the line shafting, generally one at either end, to relieve the stresses due to bending or other movement of the ship's structure.

These couplings have not been very widely adopted as many engineers fail to see the utility of them, and unless of the very best make their bearing surfaces wear rapidly and by their looseness cause considerable noise and jar when the engines are running at unequal speeds.

There is no doubt that these couplings allow the shafting to accommodate itself to any movement that may take place, in a way that could not be obtained if the couplings were held rigidly together as in the ordinary practice, but judging from the comparatively small number of ships that are fitted with flexible couplings, it would seem there was no great need for them, and probably the fault they are supposed to remedy could be reached in some less expensive manner, as the first cost of these couplings is considerable, although their maintenance is not by any means great.

Condenser.—The condenser seems to be the part of the machine that has changed least with the evolution of the present-day marine engine. The principal change has been that the condenser has been detached from the main structure of the engines and is now in many cases being carried higher up on a built-up seating, which arrangement allows for better drainage to the air pumps, easier leads for inlet and discharge piping, and also gives greater access for overhauling the main engines.

Surface condensers, which are really now the only condensers we may consider in marine work, have serious troubles from leaking and faulty tubes. Steamship owners and other authorities have spent considerable time and money in experimenting with the various kinds of tubes, but so far the results have been neither very encouraging nor satisfactory.

The trouble usually found is that after a short time in use, the insides of the tubes are found to be eaten away in spots, and pinholed, as if some active erosive action had been taking place and had attacked certain parts of the material, which is evidence enough to some minds that the tubes are not altogether homogeneous. The various methods of packing condenser tubes have undergone little change, the screwed-in gland being what is now most commonly used, but the materials used in packing the tube ends have changed frequently and have given considerable trouble.

Rubber rings, fiber rings, waxed cotton lacing, and lacing soaked in linseed oil have all been tried in turn, but experience has shown that lacing soaked in oil and properly packed in place makes the most satisfactory job and lasts well under the various changes of temperature. Wooden ferrules soaked in oil and driven into the tube plate makes a much cheaper method of packing a condenser, but has this disadvantage, that if the vessel should be laid up and the condenser remain dry for a length of time the ferrules shrink and split, and before the condenser can be made tight the ferrules require to be renewed. The packing of condensers is generally supposed to be a simple operation, not requiring any great skill or experience, but the results of hurried and imperfectly packed condensers are frequent sources of delay and trouble with feed water when at sea, and I am strongly of opinion that the greatest care and attention should be given to the operation of packing, and when the condenser is filled for testing, it should be with warm water of a temperature not less than 200 degrees Fahrenheit. Although the packing of condensers properly is of the greatest importance, the burden of the condenser trouble is without a doubt in the material of the tubes, in their inability to withstand the deterioration, which is admittedly serious, and more so now than in the past in view of the early probability of a more general use of water-tube boilers, as in that case the necessity of a pure feed water is beyond question.

Various reasons have been advanced to account for condenser troubles, the most prominent theory being the leakage of electric current from the electric generating plant on board, or the galvanic action set up by other conditions existing on board the vessel.

A further reason given is the bad quality of the stream of water sent through the tubes when the vessel is docking, or proceeding up or down a river. Dealing with the last reason first, close observation of the conditions that exist when a vessel is docking will show that great quantities of globules rise to the surface of the water, giving off sulphuretted hydrogen and other gases, the liberation of these gases being caused by the churning of the water due to the propeller working ahead and astern.

Granted the observed conditions of the water in docking, it is reasonable to suppose that even worse conditions may easily prevail in the condenser. Tracing the path of the water upon entering the ship, it is split up into fine streams in passing through the sieve-like grating on the ship's side, coming together again in the inlet pipe, then being broken up again by the action of the fan of the centrifugal pump and finally being split up again into small streams in passing through the tubes. Starting with the assumption that the water in which the vessel is lying is contaminated with sewerage and other foul matter usually found in river water near large cities, this breaking up of the water on its passage to and through the condenser in an increasingly heated state is bound to produce conditions inimicable to the life of the tubes, the worst feature being that, when the circulating pump is brought to rest, this foul and active matter remains in the tubes until the vessel leaves port, during which time I am firmly convinced most of the mischief occurs. If some simple means were adopted for washing out and draining the condenser as soon as practicable after the vessel is docked, it would amply repay for the trouble taken.

The claim that electric leakage is a cause of the deterioration in condenser tubes is one that has given rise to widely different opinions ("Corrosion of Condenser Tubes,") Trans. I. N. A., vol. xliv.) for the reason that the trouble has been quite as acute in ships without any electric generating plant, nor does it seem to make any appreciable difference whether the vessel is wired on the single or double-wired system. While able reasons have been advanced in support of the theory that the return currents in the single-wired system, or stray currents from defective insulation and connections in the double-wired systems, should be accepted as the cause of deterioration in condenser tubes. it is at present difficult to take this as being the only cause; there does not seem any reason why the electric current should single out the tubes for attack when there are other metals of almost the same compositions exposed to the same conditions and yet do not suffer to anything like the same extent.

Condenser-tube plates sometimes show a slight scoring or corrosion in more or less vertical lines, but that is always on the steam side and seldom amounts to anything. and is generally attributed to the acids set free from the grease brought in with the exhaust steam. The composition of the material of the tubes seems to have little bearing on the subject, as they nearly all seem

to suffer alike. Of course it is recognized that, although an electric current of itself may not attack the tubes, the effect of an electrical current passing through certain liquids is to split them up and set free corrosive elements from what may otherwise be a comparatively harmless liquid.

If we consider the conditions of the water in the engine room and stoke-hold bilges, which is a solution of fatty acids, fresh water, salt water, drainage from hot ashes and all other sources, and in which the various metals are brought in contact, it is reasonable to expect some galvanic action to take place on the fittings so exposed, and such has been found to be the case to quite a considerable extent in some instances. In some ships corrosion in copper and other pipes in the bilges has been traced to the use of certain animal or vegetable oils, but up to the present little definite progress has been made in determining the cause of the trouble in condensers (Trans. I. N. A., vol. xlv., "Corrosion in Metal Pipes"), and although further investigation may disclose causes not yet ascertained, the theory of electric leakage from the generating current cannot of itself at present be accepted as the only cause of condenser trouble, but it is conceivable that, with a

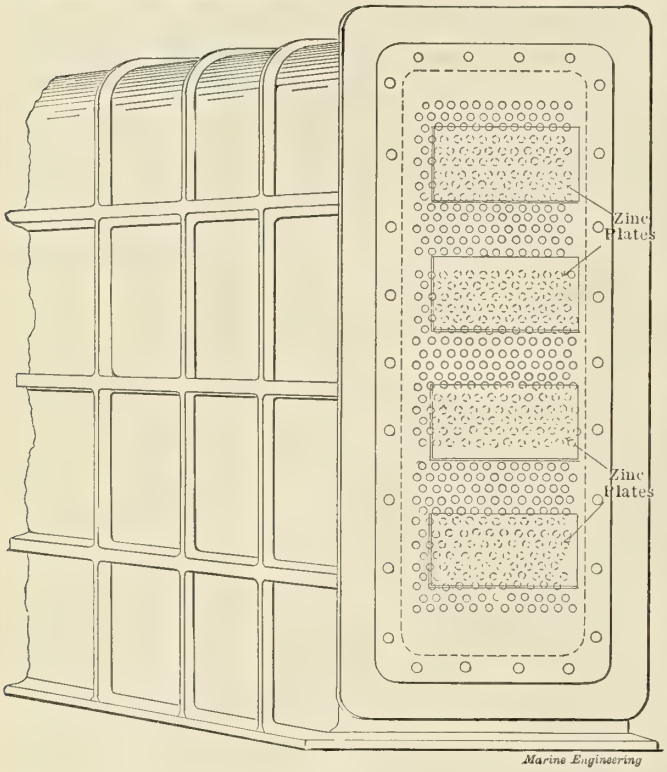


FIG. 2.

condenser having water lying in it highly charged with decomposed matter, conditions could be produced from a combination of causes resulting in action on the material of the tubes. Whether electric leakage is the cause or only a contributory cause, the fact remains that condenser troubles have become increasingly frequent since electricity came into more general use on board ships.

If it can be established that condenser troubles are due to electric current, generated from any source whatever outside of the condenser, it does not appear that it would be an impossible task to insulate the condenser in some manner and so protect it from such attacks. The use of oil in cylinder lubrication that may be carried into the condenser with the exhaust steam has little connection with the question, as mineral oils are now almost exclusively used, and further, the defect is most always found to proceed from the interior of the tubes. The working conditions of condensers have not changed proportionately with the shortened life of the tubes, and it is contended that if one part of the tube remains good, there is every reason to expect the whole tube to remain uniformly good if the process of manufacture is all that could be desired. Homogeneity is a word much believed in by metallurgists and engineers, but, after all, it is only a comparative term and has wide definitions. It is admitted in the very fullest sense that the tube makers have given this matter very considerable attention, and it is hoped they will continue to do so, as there is still something to be done in the way of producing a tube that will come up to the requirements and so rid shipowners of what has been a source of vexation and delay, and also a costly item of maintenance.

Numerous devices have been adopted to counteract the deterioration in the tubes, and the most successful has been to line the water ends of the condensers with rolled zinc plates on studs fitted in the tube plates (Fig. 2). This has been found to improve conditions quite considerably, but is not by any means a panacea for the ills that condensers are heirs to and which will be arrived at only by continued and persistent effort.

Boilers.—In dealing with the present high pressure Scotch boiler, it will be generally agreed that it has now reached its limit in size, thickness of plating and pressure, and although a few boilers of reduced diameter have been built for pressures up to 267 pounds per square inch, the pressure that is found in most general use in our latest boats ranges from 200 to 215 pounds per square inch. The maintenance of boilers with the higher pressures now in use has not materially altered from the conditions existing when lower pressures were dealt with, so long as the conditions of combustion are the same, or, in other words, with natural draught; but as a large proportion of ships are now fitted with assisted draught of one type or another, it is considered desirable to discuss the maintenance of boilers under forced draught conditions. Experience has undoubtedly shown that the working conditions of boilers under forced draught can be made radically different from the conditions which exist under natural draught. The question of forced draught will be dealt with later on; at present it is proposed to discuss only the effects on the boilers. In natural draught we have the temperatures in the combustion chambers due to energetic working of the fires and the draught produced by the funnel, which at no time becomes so excessive as to cause damage to the chamber plates, unless the boilers are allowed to become very dirty from scale on the water side.

With forced draught we have temperatures in the furnaces and chambers of much greater intensity, and which are limited only by the amount of coal capable of being burned on the fire grate and the air pressure obtained in the ash pits, with the result that the combustion chamber plating becomes locally overheated and buckling takes place; stay nuts are burned off, tube ends leak, and the plating cracks through the landing edges of the rivet holes, and this is especially severe in places where any broad landings or doublings are exposed to the flames. Various means have been adopted to protect the surfaces at the back ends, mostly affected by the flame, by fitting tiles made of fire clay or other refractory material and molded to the shape required, but their usefulness is sometimes of short duration, owing to the difficulty of keeping them in place, as the means taken to secure them are rapidly burned off. The most serious trouble met with is in the corners of the furnace plating, where it is flanged up to take the back tube plate, and here, unless the radius of the flanging of the tube plate has been of an easy nature, cracking very soon develops, and as this takes place through a riveted connection, to repair which a three-ply thickness of material will be necessary in places, and which it is found is an impossibility to keep tight in boilers under forced draught conditions. It is true this weakness has been somewhat modified by the adoption of the furnace with the "Gourlay" neck (Fig. 3), but even with this improvement the difficulty experienced in keeping this joint tight is quite acute, as the riveting and calking requires frequent renewal, and each time it becomes more serious than before, by reason of the deterioration of the material causing cracking through the rivet holes and landings.

That this weakness is a common experience can be abundantly proven. Where a boiler is subjected to a high air pressure for a length of time the invariable result is distortion, cracking, and leakage.

The radius of the flanging at the sides of the tube plates to which the furnace back end is riveted should be sufficiently great to allow the plating to be properly laid up and a good riveted connection made without resorting to the use of tap bolts and other such makeshifts.

The other most frequent weakness is shown in the riveting of the furnace to the back tube plate, right over the furnace crown. (Fig. 4.)

This joint is in many cases double riveted, and with two thicknesses of material of a breadth sufficient for a double-riveted joint exposed to the hottest part of the flame, it is found extremely difficult to keep it tight, as the part of the plating exposed becomes overheated and weakened and the movement resulting from this overheating gives rise to extensive leakage. A single-riveted joint has been found to stand perfectly well at this part, and there is no good reason why it should not do so if the spacing of the riveting is properly designed and the work is of the best. It is at this part that protection as previously referred to by firebrick blocks is urgently required, and when the voyage is short and the blocks can be frequently renewed, they are an undoubted benefit.

In the S. S. *Finland* the builders had the whole of the combustion chambers of one boiler welded throughout, the only riveting being the attachment of the furnace to the back-tube plates.

This was found to be a highly satisfactory job and has not, I understand, given any trouble whatever, the increased cost, I think, being the only objectionable feature.

In the case of the nuts of staybolts being burned off, this has been met by doing away with the nuts on the stay ends that come in a direct line with the furnace, and reverting back to the old practice of riveting the stay ends over, which has been found to answer very well. As the maintenance of boilers has become a serious consideration, it cannot be too strongly pointed out that experience and common sense are just as essential in designing a boiler as it is to get the maximum amount of heating surface into the minimum size of shell. Many boilers are so designed that they are crowded, too closely pitched stays in the steam space, tubes brought close out to the wings and down on the top of the furnaces, with a result that the boiler cannot be either properly cleaned or examined, circulation is retarded, priming is common, and altogether the working conditions of the boiler are unsatisfactory; and further, where such conditions exist, corrosion is always active.

If we consider the various elements that go to make up the ordinary cylindrical or Scotch boiler, it will be recognized that we have a mass of unequal stresses to be taken care of, each stress tending to a movement in the structure and each movement meaning so much fatigue to the material, which in time culminates in leakage and possibly rupture.

The evidences of distortion most commonly met with are:

(1) Cracking set up in the flanging of the back tube plate and furnace connections at this part.

(2) Cracking in the furnaces at the line of fire bars in a longitudinal direction.

(3) Cracking on the knuckle of the flanging of the lower front heads, and also in the case of double-ended boilers the tendency to rupture of the plating through the line of rivets in the circumferential seams at the bottom of the boiler.

(1) This is caused by the unequal expansion of the furnace and the tube plate, due to the upper half of the furnace being so much hotter than the lower half, and also to the impact of the flame on the tube plate joint where there are two thicknesses of material which does not allow of the heat being quickly absorbed by the water.

It is fair to assume that in the case of the back tube plate, perforated with holes and unequally held as it is, there is considerable distortion, as is shown by the leaky tube ends and riveted connections, and also not infrequently by cracked tube plates.

This movement, however, may not be altogether due to the change of form in the tube plate alone, but is doubtless assisted by the action of the furnaces under the varying temperatures to which they are continually subjected, but that the movement is positive and actual has been demonstrated beyond a doubt, both under working conditions and by careful experiment. Mr. Yarrow conducted a series of experiments in 1891 (Trans. I. N. A., vol. xxxii., p. 98), and showed conclusively that with every change in temperature the conditions of the tube plate and connections changed proportionately. At a later date, Mr. Milton, Chief Engineer Surveyor to Lloyd's Register, submitted the results of a considerable number of tests (Trans. I. N. A., vol. xxxiv., p. 157), made by hydrostatic pressure on the various types of marine boilers, and in every case movement of the boiler took place when under pressure.

A case occurred quite recently, with one of the most eminent and best equipped builders in this country, when the question arose as to whether any vertical movement of the combustion chambers took place when under pressure, it being contended that if no movement took place, it would not be necessary to stay the chamber bottoms to the shell.

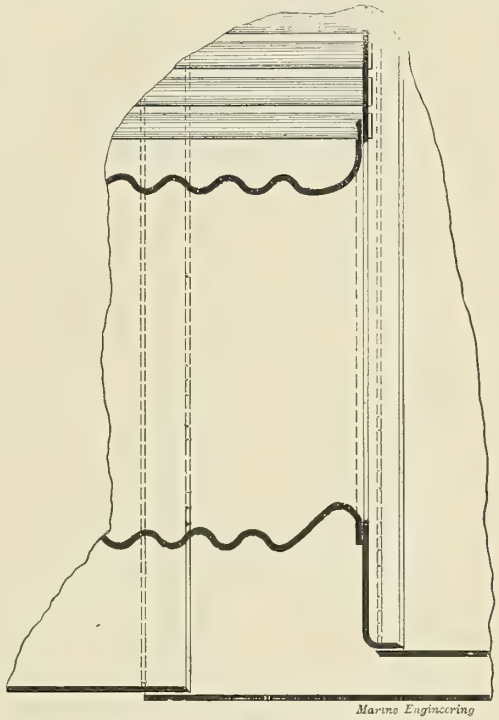


FIG. 3.

The boiler was an ordinary double-ended, high-pressure boiler. To take the actual measurements a hole was drilled in the bottom of the shell into which a small water-tight gland was fitted with a rod passing through and abutting against the bottom of the middle chamber. The rod was carefully marked when the boiler was completely full of water, but without any pressure. A steady pressure was then applied from accumulators on the premises, and at 40 percent in excess of the working pressure the test was stopped, as it was found the distance between the bottom of the shell and the chamber bottom had increased 7-16 of an inch. How much the movement would have been if the pressure had been allowed to go up to twice the working pressure can only be assumed; perhaps some of the wing stays would have given out, but the builders wisely accepted the results and decided to fit the stays.

Further evidence has been obtained of like results under actual working conditions.

The chief engineer of one of the large mail boats sailing out of this port furnished me with data showing that if the cleaning of a fire could be kept within a certain number of minutes no bad effects to the tube plate and furnace joints were observable; but when from dirty fires or other reasons the operation was pro-

tracted beyond the stated time leakage would commence and continue until the fire was well under way again. He also informed me, and this I personally observed, that where this leakage had taken place, careful inspection showed that the calking had been sprung by the lowered temperature, causing local contraction in the furnace and combustion-chamber connections.

The experiment was tried by putting two men to clean a fire so as to reduce the time required, and the results showed beyond a shadow of doubt that the leakage was directly due and proportional to the length of time the furnace door was open.

(2) This is usually observed after the furnaces have been in use for some time, and is generally found to develop about the middle of the length of the furnace in small hair-like cracks showing up on the points of the corrugations nearest to the fire. These faint cracks steadily increase in length and extend into one another until they become so bad that they require to be pinned or the furnace removed altogether.

The cracks in the furnaces are undoubted evidences of fatigue, produced from the stresses set up in a vertical direction by the unequal expansion and continuous movement due to the change of temperature between the upper and lower halves of the furnace and localized along the line of fire bars where the furnace is weakest, owing to the intense heat and excessive wear at this part. Cracks of a more serious nature sometimes develop in a circumferential direction in furnaces; this is not due to fatigue, but rather to carelessness in manipulation of the boilers.

A case came under the writer's notice some time ago where in a vessel the morning after arrival in port, upon the engineer's proceeding to examine one of the main boilers, that had been blown down 24 hours previously, they discovered a crack 20 degrees long in a circumferential direction in one of the lower furnaces. The cracked furnace was temporarily repaired, and upon the vessel's return to this country two months afterwards a crack of similar size was found under precisely the same conditions in one of the other boilers.

Inquiry elicited the fact that it was the practice to lay off for cleaning purposes one of the boilers in rotation, while the vessel was in port on this side, the *modus operandi* being to draw fires, blow down the boiler and drop in the manhole doors as soon as practicable after the vessel was made fast in dock.

It was pointed out to the engineer that this mishap was altogether preventable and was due to a lack of appreciation of the care necessary in handling boilers. The conditions were simply that the boiler in a limited space of time had all the heat and internal pressure removed and the empty boiler was then exposed to the chilling effect of a 36-inch down-take ventilator discharging in front of the boiler air of a temperature near to freezing point, as it so happened that it was in the winter months that the damage occurred.

Arrangements were made that in the future, upon the fires being drawn, the furnace door and ash-pit dampers should be closed, the ventilators turned back to wind, and the water left in the boilers to be pumped out next day, and since this has been done no further trouble with the furnaces has been experienced, clearly demonstrating how urgent the necessity is for a better appreciation of the care requisite in dealing with boilers.

(3) Cracking in the knuckle of the flanging of the lower front heads and distortion of the boiler bottoms is in some cases a serious trouble from the very start, due in a great measure to design. These stresses at these parts are very severe indeed, in some cases causing rupture of the shell plating through the line of circumferential riveting in the bottom of the shell, and cracking along the knuckle of the flanged part in the lower head. Investigation has demonstrated that to the differences of temperature between the top and the bottom of the boiler, causing unequal expansion of the shell plating and boiler ends, is due the distortion previously mentioned. Leakage at the boiler bottoms is difficult to cure, as heavy plating and rivets cannot be efficiently laid up by hand, and least of all in a confined space under a boiler bottom.

The flanging of the lower heads when examined inside show fatigue markings similar to the fine cracks shown on the fire side of the furnaces, and these in time develop upon the outside of the flanging also, and becomes so serious as to require the lower head to be repaired or renewed.

This cracking is found to develop at an early date in boilers where the flanging is of a small radius, showing that the plate in its efforts to accommodate itself to the changes of temperature and the movement resulting thereupon had been distressed, and of course where the material is fatigued to this extent there is a known limit to its endurance.

The lower heads also suffer from wasting caused by the water draining from the hot ashes corroding the fronts of the boilers. This, however, is to a great degree preventable and can be taken care of.

The lower heads as a result of the excessive flanging done upon them are peculiarly susceptible to corrosion on the knuckle of the flanging, and where this wasting is allowed to go on fatigue of the material follows very rapidly, and where rupture takes place at this part it is a difficult and costly business to repair, coupled with the fact that it can never be kept tight.

Boiler making has become a science of considerable exactitude, and with the excellent materials and tools now in use there can be no excuse for bad boiler work.

If boiler shells were ideally circular and could be kept so, if the flat surfaces could be so stayed that the various forces were balanced at all stages, then ideal conditions would have been obtained; but as the shell is distorted from its true form by various internal stresses, such as staying compensation for hole cutting, etc., the end plates which are so widely dissimilar in the areas under pressure that the staying can only approximately meet the stresses, show, and that very forcibly, how vitally important it is that all the possible conditions should be carefully thought out and amply provided for in a boiler design.

It cannot be too strongly urged how desirable it is that boilers should be kept free from scale and dirt.

Incrustation and corrosion go hand in hand. It is seldom that scale is removed without disclosing oxidation.

Stromeyer estimates that scale offers one hundred times as much resistance to the passage of heat as an ordinary steel plate of the same thickness, while a film of grease shows a resistance ten times greater even than scale, and as scale found in marine boilers always contains a certain proportion of grease, it is at once evident how serious the deterioration in a boiler may become when working under these conditions.

Good design, or the best class of workmanship, count for little if the boilers are allowed to get dirty and form scale, plating bulges, furnaces collapse, tubes leak, and all from causes that care and attention would in most cases have obviated.

At present many owners consider it cheaper wherever possible to use fresh water altogether for the boilers, rather than meet higher coal bills and the expense of scaling. Grease deposits in boilers have been to a large extent eliminated by the use of filters, and also from the disuse of cylinder and piston-rod lubrication to a large extent. Of course, it is well known that feed filters, however efficient, can arrest only the solid matter contained in the feed water, leaving whatever may be carried into the boilers in solution to be dealt with by zinc plates and other detergents which are used in boilers for this purpose.

The boiler question could not be dropped at this stage without considering the suitability and maintenance of water-tube boilers in merchant ships. The boilers in merchant ships must be capable of continuous working at the maximum. Reduced weight can be only of relative importance, except in very large installations, and should never be accepted at the sacrifice of reliability.

A brief examination shows that the principal claims made on behalf of the water-tube boiler are:

(1) Reduced weight per square foot fire grate.

(2) Reduced weight of water per H. P. developed.

(3) Greater adaptability for higher pressures without increasing weight.

(4) Greater facility for raising steam without injury to the boiler.

(5) Greater facility for repairs or renewals without disturbance of the ship's internal structure.

It is not proposed to examine or discuss the above points from the theoretical or constructional side, as that has been dealt with ably by Admiral Melville in the proceedings of this Society, and many other writers elsewhere, but rather to deal with the question of the suitability of water-tube boilers taking the place of the present shell boiler in merchant ships large and small.

(1) The difference of weight alone will not have much influence with the owner of the smaller vessels, such as the average tramp steamer with two or three boilers, as he has to consider the fact that his stoke-hold crew, and to a greater or lesser extent his engineers also, are continually changing, and that the vessel may at any time be in a port where the materials for the repair of water-tube boilers could not easily be had, and, beyond all, the necessity that he should provide ample means for a fresh-water feed make-up to meet the losses which, though common in all boilers, experience has shown are certainly not less in water-tube boilers.

In large installations the question of weight becomes of greater importance, as in the case of a ship where the boiler-room weights are, say, 1,200 tons, the reduction in weight is considerable in view of the builders' claims that the saving in favor of water-tube boilers is not less than 30 percent. This coupled with the very much greater saving in weight of water carried in the boilers are all-important factors to the naval architect and engineer, who are striving to produce a certain tonnage and horse-power on the least safe weight.

Everything else being equal the desire of the ship owner to save weight is just as keen as the builders, but it is here that the dividing line in engineers' minds is so sharply drawn, as it is contended by many that the saving of weight by the adoption of water-tube boilers is at the sacrifice of continuous and reliable working.

It is not upon the failures of the past that the water-tube boiler is to be condemned, but rather on the present and the prospects for the future that it is to be judged. There have undoubtedly been many bad water-tube boilers built, but there have certainly been plenty of bad shell boilers built also, and the present-day shell boiler, which is the outcome of long experience, bears little resemblance to the article that was produced forty years ago.

What the water-tube boiler has suffered from largely in the past has been an unripe advocacy of their use; the boilers have been installed in ships where the conditions under which they had to work were as little understood by the builders as the new type of boilers were by the engineers, the consequence being that in many cases the boilers failed utterly and were taken out, and it was only a courageous owner who would make a second venture in water-tube boilers.

The water-tube boiler of the present day bears little relation to that of even ten years ago. In the construction of this type of boiler at present castings have been altogether eliminated, and the highest quality of steel is now used for all purposes entering into construction, and this in combination with the superior tools now used in this class of work has resulted in producing a generator the workmanship of which is excellent. The question as to which type of boiler is most suitable for merchant ships, the maintenance of which shall be normal and which shall be capable of working at its maximum for lengthened voyages, has been engaging the attention of engineers all the world over for the last five years, to a greater extent than it has for the past twenty years, and the opinion is steadily but as surely growing, that given the proper type of boiler for the specific purpose in view, and due provision made for meeting the new conditions, there exists no well-defined reason why the water-tube boiler should not be successful. The British Boiler Commission, that has just completed its labors, has, after most extensive tests, unhesitatingly decided in favor of water-tube boilers.

It has not always been the fault of the boiler that has led to condemnation, but more often on the part of the engineering complement, both afloat and ashore, to thoroughly appreciate and understand the new conditions involved, nervousness and lack of patience contributing in no small measure to disaster.

(2) The reduced quantity of water carried in water-tube boilers is to a considerable extent offset by the necessity of an increased quantity of fresh water carried in the double bottom for make-up feed, and experience has shown that the percentage of make-up feed required in water-tube boilers is much higher than in the case of shell boilers. Some boilers have been tried with partly salt water, but I do not think engineers are yet prepared to adopt that course, but rather to get as pure a feed as ever possible.

Water-tube boilers mean the installation of evaporators of greater capacity, unless the voyage is short, as ships cannot afford to carry large quantities of fresh water and so shut out cargo. The greatly reduced quantity of water carried in the boilers is

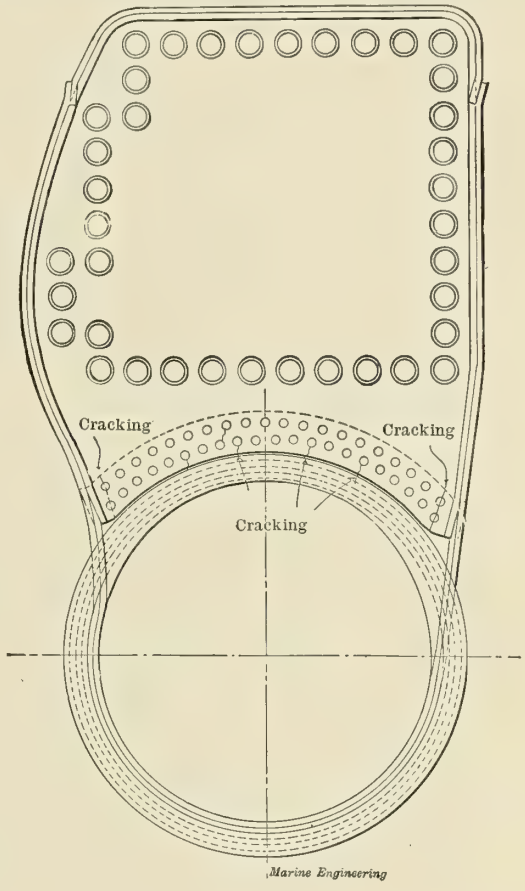


FIG. 4.

from one point of view a drawback, as the margin of safety between "water" and "no water in the glass" is very materially reduced. In a shell boiler the loss of water in the glass from priming or any other cause is not immediately serious, and the boilers can usually be handled with safety until the water is restored, but in the case of the water-tube boiler the loss of water in the glass is instantly dangerous.

It may be a simple matter to figure out what length of time may elapse from the time the water disappears out of the glass until the boiler is in jeopardy, but few engineers are prepared to take such risks, in the ordinary pressure of stokehold work, as a boiler failing to show water is an unknown quantity and should be put out of active use to prevent mishap. This danger from shortness of water in the large tube type is not so acute as it may be in the small tube or express boiler, but in any case the slightest derangement of the feed becomes serious immediately.

Complicated feed arrangements have in the past been altogether too prevalent with water-tube boilers, and have caused a corresponding amount of trouble and damage, but arrangements of greater simplicity are now being fitted, and there is every reason to expect the feed arrangement of water-tube boilers to be just as simple as in the ordinary shell boiler.

(3) There is no question of doubt as to the flexibility of water-tube boilers, when it comes to the question of higher pressures, without increasing the weight to any material degree, as the steam drum is the only member requiring to be considered. This factor is only of comparative value at present, as the most recent development hardly indicates that higher pressures are being seriously considered, the tendency being to obtain greater economy by other arrangements.

(4) With shell boilers, where well thought out, and feed and boiler circulating arrangements have been adopted, there is little likelihood of damage by raising steam quickly. When the pipe arrangement is so designed that the pump can draw water from the bottom of the boiler and circulate it back through the feed heater at boiling point by the feed check into the boiler, the expansion of the boiler goes on gradually and easily, and fires can be started and steam raised in a few hours without any resulting damage. It is, however, not desirable to raise steam too quickly, nor is it often found necessary in merchant ships to do so, as they usually have scheduled times for sailing or they are so situated that due notice can be given.

In ferryboats, where the vessel is kept closely at work or in ships other than merchant ships, this factor is no doubt of great value, but in merchant ships it can be of only minor importance.

(5) The necessity of making provision to allow for renewal of a ship's boilers is never considered during construction, although it has to be faced in many ships during their life time, and is always a costly and tedious business, involving much loss of time and damage to the ship's internal structure.

Where the opportunity has admitted of boilers being well taken care of and a long life of work of upwards of twenty years has been gotten out of them, many owners do not consider it a good investment to reboiler a more or less obsolete ship, as the extra expense due to reboilering is very considerable in old ships for the renewal of decks, tank-tops, bunkers, etc.

Where boilers require to be renewed, the fact that the water-tube boiler can be split up into its separate members and passed either through the funnel opening or through a bulkhead is of undoubted advantage to the owner, but I am convinced that the two points (1) and (2) are the all-important questions that will be considered by engineers and others as to what type of boiler is suitable for merchant ships.

There is not, in my opinion, any sound mechanical reason why water-tube boilers should not work well—enterprise and patience will demonstrate this assertion—experience and confidence on the part of the personnel will be gained, better baffling will result in an improved efficiency and economy, just as spiral retarders have improved the economy of the shell boiler, and as the boilers come into more general use they will be better understood, and will, without doubt, be improved, and become as reliable as any other type of boiler.

The successful working of water-tube boilers lies in the more highly capable and intelligent management of the stokehold.

Forced Draught.—Forced draught has certainly been a most valuable development in engineering, the proof of which need not be presented here; the main question arising in this paper being the effect that forced or accelerated draught has had upon the life and maintenance of boilers.

The two systems to be found in merchant ships are "Howden's Forced Draught" and "Ellis and Eaves Induced Draught," each of which in competent hands is capable of working the boilers at a very high efficiency. The closed stokehold system has confined itself largely to war ships and will not be considered here.

The effect of forced draught upon some boilers has been very marked indeed, and the question is of far-reaching importance as to what is the best air pressure to be carried in the ash pits. In some vessels 1-2 inch is considered sufficient, and is then really an assisted draught, while in others pressures up to about 3 inches are carried in the ash pits. It would almost seem as if somewhere between those two extremes would be found the desirable mean, but after close observation I am firmly of opinion that in shell boilers at all events the pressure in the ash pits should not exceed 1 inch of water at any time. I have examined many boilers where high air pressures have been in use, and the results have always been the same—buckled and cracked plating, stay-nuts burned off, calking started, and leaky tubes, whereas, when the air pressure was not allowed to exceed 1 inch the condition of the boilers was as good as those working under natural draught.

The maintenance of boilers under the higher air pressures is altogether out of proportion to the extra work got out of them, with the further result that a continuance of these higher pressures will cut out not less than 30 percent of the useful life of the boiler. The further objection to high air pressures, and it is a potent one, is that there can be no attempt at coal economy with excessive pressures, as experiments and trials without number have fully demonstrated. The temptation is keen to use higher air pressure when making up time, when coal is bad, or in the endeavor to get a power out of the boilers that their size does not warrant.

These remarks can be applied to water-tube boilers also, although their physical condition is not so seriously affected as the shell boiler; but experience will show that even there a moderate air pressure in point of maintenance and coal consumption will be found more than desirable.

Oil Fuel.—The maintenance of boilers working with oil fuel is, at the present time, of decided interest, in view of the possible development in this direction. It is not proposed to deal with the question of oil as a fuel, but rather to consider its adaptability and its effect upon boilers. The use of oil fuel has been with a few exceptions confined to oil-carrying steamers, but there is not any structural reason to debar its use in ordinary cargo or passenger ships, as the fuel can be carried in the double bottom and in tanks fitted in the usual coal-bunker spaces. With proper precautions taken the risk of explosion is very remote, as the flash point of the fuel is usually not less than 150 degrees Fahrenheit.

When the fuel is carried in the double bottom a heavy layer of cement 3 inches thick, properly sectioned off to keep it in place, has proved to be a good insulator for the tank.

The two systems most widely applied to burning oil fuel are compressed air or spraying by steam.

The former requires a small air compressor, or other suitable means for obtaining the required pressure, which is usually about 1 inch; the other method, spraying by steam, requires the installation of extra evaporators to make up the loss of water used in spraying the oil, which is found to be about 3 percent of the total steam evaporated. With the compressed air a certain amount of steam is, of course, required to drive the air compressor, but there is no doubt that with the air heated to a high temperature this process will be found to show a high efficiency, especially in long voyages, as under oil-fuel conditions the combustion and the steam are steady, which always means economy.

The condition of the boilers after using oil fuel is found, when the necessary safeguards have been taken to protect the surfaces of the combustion chamber from direct impact with the flame, not to differ to any material extent from that of boilers burning coal.

The combustion chamber backs and sides require to be protected with brick work carried up above the level of the furnace crown, and in most cases the furnaces are bricked over also, and arrangements of brick arches erected in the furnace to baffle or break up the torch-shaped flame within the length of the furnace. When these conditions are properly carried out the boilers show not the slightest sign of distress, but if the brick work is allowed to come down it means that the flame strikes straight through into the chamber; the furnace does comparatively little work,

intense heat is localized in the combustion-chamber back and tube plates, causing serious distortion and leakage.

I have examined boilers that were less than twelve months in use, where the combustion chambers were seriously damaged from neglect to keep the brick work in good condition. The burning of oil fuel is supposed to be a simple operation, and so it is when we know how to do it.

There is every reason why boilers should work well under oil fuel. The temperature of the gases of combustion varies but little; there is no opening of furnace doors and cleaning fires; the men readily handle it intelligently, and the maintenance of boilers working under oil fuel should, in capable hands, be less than under coal conditions. The drawback at present to its wider use seems to be more commercial than mechanical.

Superheating.—The interest now being taken in the question of superheat seems to indicate that we have arrived at another period in the cycle of development in engineering.

When steam pressures ranged from 40 pounds to 80 pounds superheaters of many types were in general use, and were considered almost indispensable, and at these low pressures they certainly were of value; but with the advent of the triple- and later the quadruple-expansion engines, superheaters went out of use.

The whole aim of engineers for some considerable time past has been to seek for the improved economy in higher steam pressures, the quadruple following the triple when pressures rose above 180 pounds per square inch, and later, in the case of the *Inchdune* and other ships for the same firm, the five-crank engine was developed with Scotch boilers working at a pressure of 267 pounds per square inch.

Engineers are now convinced that if any further economy is to be reached by higher pressures, a very radical departure will require to be made from the present type of engine and boiler. The times would seem to be propitious for this change, as shadowed forth in the development of the steam turbine and the more general use of water-tube boilers, but even in the turbine it is recognized more fully than in the reciprocating engine that there are great possibilities for a higher efficiency and economy by the use of superheated steam. Dealing with the present reciprocating engine, it is well known that the moisture or water particles in the steam act as a lubricant for the leakage of the steam between the piston and the cylinder walls; this leakage being excessive in the case of many piston valves; and as piston valves by reason of their equilibrium under working conditions are now very widely used, it is recognized that with steam of an increased volume and more closely approaching a gas, leakage will be less likely to take place and efficiency will be improved.

It is not proposed in this paper to deal with the development of the steam turbine, but reference may be made to the fact that superheating can with perfect safety be carried to a much greater extent and with corresponding advantages in the turbine than would be possible in the reciprocating engine. In the turbine there are no surfaces such as cylinder walls, valve faces, piston rods, etc., so that no limitations to the amount of superheating will be met with in the machine. In the reciprocating engine it will be very different, as there very high dry temperatures may readily cause abrasion of the surfaces; but it is not considered that from 100 to 150 degrees Fahrenheit of superheating will produce conditions that cannot easily be dealt with. It is in the multiple method of superheating wherein the greatest possibilities seem to lie; that is, in superheating the steam on its way from the boilers to the engine, and then after doing work in the cylinder it is passed through another heater, and so on to each succeeding cylinder, the steam being superheated in each change. This no doubt means a certain amount of complication, and perhaps weight, but as it must increase the efficiency of the steam quite considerably the extra weight in superheaters, it is claimed, will be equalized by the reduced boiler weights required. If superheating be applied to the initial stages alone, a decided gain would result by getting rid of the water brought over by the steam from the boilers by priming or other causes, and would result in lessened leakages at the engine and get useful work out of a certain proportion of the steam that would otherwise, by condensation, become inert. Of course there are many objections to superheaters which are well known and need not be recapitulated here, but there is not the slightest doubt that with the renewed interest now being taken in this subject, superheaters of a widely different type from those of the past will be developed.

There would, of course, be a certain percentage of drop in the pressure, due to the friction and wire drawing through the passages and connections of the superheater, but with well-designed apparatus this drop should be only small, and would be more than compensated for in the higher quality of the steam.

The whole question of superheat is likely to lead to considerable development in the early future if engineers are able to overcome the objections of the ship owners to what many consider only an added complication.

Main and Auxiliary Steam Pipes.—The necessity for a good pipe arrangement was never greater than at the present time. As stokeholds extend in length the demand for well-designed main steam pipe lines is very urgent, so that the losses from drop in pressure between the boilers and engines should be kept as low as possible. As long lines of steam pipes have considerable movement when under steam, the greatest care should be taken to avoid any abrupt changes in the direction of the line of piping, and where it is found necessary to make changes, expansion joints should be fitted and so arranged that it will not be possible to subject the bends and other connections to any stress beyond that due to the boiler pressure. Main steam-pipe flanges should not be bolted to bulkheads unless expansion joints are in the immediate vicinity, and so arranged to take up any movement in the bulkhead that may take place; and never under any consideration when struts carried from any other part of the machinery are attached to the bulkhead in proximity to the steam pipes, as grave danger is to be apprehended in this arrangement in the event of the vessel grounding or meeting with other serious damage. Long lines of steam piping mean considerable condensation, and ample means should be taken to ensure this water being trapped, and where this is done automatically hand drains should also be fitted, so that in the event of any of the lines being laid off temporarily, the engineer will always be able to inform himself as to whether the pipes are clear of water before turning steam on.

The arrangement of auxiliary steam lines in many cases leaves much to be desired; pipes are dropped down to low levels to get round some obstructions, then raised abruptly to higher levels, thus forming a pocket which is a lodgment for water, and besides being a positive danger, it lowers very materially the efficiency of the whole auxiliary outfit.

I am perfectly familiar with the fact that all pipe arrangements are a compromise from beginning to end, the ship's structure and other considerations requiring to be dealt with at every step, but I am convinced that where careful attention is given to the detail of the whole arrangement much greater satisfaction will be obtained, and the wear and tear on the steam end of the auxiliaries will be correspondingly reduced. Few auxiliaries can lay claim to a very high steam efficiency, this being due in no small measure to the quality of steam with which they are operated.

It is not possible to lay down any lines upon which a pipe arrangement should be designed, the exigencies of each case requiring special consideration, but the chief aim should be to avoid complications; the use of cast iron in any form should be forbidden, copper bends and expansion arrangements being fitted to accommodate the movement that takes place in the long stretches of auxiliary steam piping.

Pumps.—One of the greatest advances made in the development of the present triple or quadruple engine has been in relieving the main engine of the duty of driving the pumps and using the power for propelling the ship only.

The long-stroke, slow-working simplex feed pumps now in use, when fitted with automatic speed regulators, give good results;

the feed line is not subjected to shocks and as the pump works only when there is water to pump, it means that less air is forced into the boiler along with the water. As the water ends of high-class feed pumps are of brass, the valve seats should be cast solid with the chamber; removable seatings that are only "driven in" should not be accepted, as they are sure to become loose unless they are well secured in place by screwed pins through the flanges into the chamber, and as these pumps require to pump water that is well over 200 degrees Fahrenheit against a high pressure, this class of pump requires to be of substantial make and the best workmanship; otherwise they are a continual source of trouble.

One other good feature of the independent feed pump, especially in large installations, is that they are usually fitted in pairs and interchangeable, which means that the breakdown of a feed pump is not so serious as formerly, and as most pumps of this class have considerable reserve capacity any mishap occurring to put a pump out of action does not affect the speed of the vessel.

The whole array of auxiliary or independent machines, such as feed, bilge, ballast, air, circulating and sanitary pumps, electric light, blower and ventilating fan engines, etc., now usually fitted, are an elaboration of the idea of separate units for each particular service, and have in most cases been found to do their work remarkably well; but it has to be remembered that the maintenance of such a large number of steam users, in even a moderate degree of efficiency, entails a large amount of work and expense, and as many of these auxiliaries are so placed that examination and repair are difficult to accomplish, it can be easily understood that they are neglected and become most uneconomical steam users. This feature of auxiliaries is perhaps not much considered, but investigation will disclose what a surprisingly large proportion of the boiler power is absorbed in driving the auxiliary machinery.

Electric power in some cases has been made use of for driving quick-moving auxiliary machinery, such as blower and ventilating fans, centrifugal pumps, etc., and that there is a wide field for electric power in this direction is undoubted, for although the first cost of the electric installation is greater the up-keep and losses are much less. There is at present a certain amount of distrust in the minds of many engineers against electric motors taking the place of the steam engine when a positive action is demanded. That there is considerable force in this objection will be realized in the event of the motor attached to the circulating pump breaking down when leaving the dock, and bringing the engines up from lack of vacuum, and there is this further danger, that, as circulators are carried low down, an excess of water in the engine room might very easily put the motor out of action altogether. Notwithstanding the objections against the use of the motor, I firmly believe that their use will become general where continuous running is required, and if the motors are properly cared for and not overloaded, satisfactory results will be obtained. Cross-connections are always arranged for to provide against such contingencies as the breakdown of the circulator, but unless electricians can produce a motor, proof against the effects of oil and salt water, circulators should not be electrically equipped.

Small steam turbines, for which a fair economy is now claimed, seem well adapted for high-speed continuous running; their action is positive, and the wear and tear is practically *nil.* The whole question of the adaptability and location of auxiliary machinery is now receiving very much better attention than hitherto, and rightly so, as a bad arrangement of this very important factor is a serious drawback; the up-keep is excessive, breakdowns are continuous, and are most harrassing to the whole engineering staff, as it means continual toil from port to port, and all from causes that experience might well have prevented.

The question as to the best steam pressure, in point of economy and efficiency, for operating auxiliaries, is of decided interest. It does seem an anomaly to raise steam up to the high pressures in use at present, about 210 pounds, and then wire-draw it through a reducing valve down to 100 pounds to make use of it.

Few auxiliaries are well adapted or kept in such mechanical efficiency to profitably use steam of the higher pressures, as the losses from radiation and condensation and other leakages are considerable, and as auxiliaries are not an afterthought, it would undoubtedly seem to be a better arrangement for a certain proportion of the boiler power to be designed for the lower pressure. The first cost would be decidedly less, as the price of boiler power and all the necessary fittings for 100 pounds steam pressure is very much lower than it would be for 210 pounds. Some vessels have been arranged along these lines and the results have been very satisfactory indeed.

GENERAL.

There are many other points that might have been incorporated in this paper, such as feed heating, use of evaporators, filters, etc., but enough has been dealt with to show that the efficient maintenance of machinery at the present day is a living complex reality, and demands from those who have to take up the responsibility a high order of intelligence and unremitting care.

The later developments have unquestionably demonstrated that rule of thumb methods are no longer admissible, and where such valuable property is at stake the best in design, material, workmanship and supervision is demanded, and only the best should be accepted.

The Engines of the English Armored Cruiser Argyll.

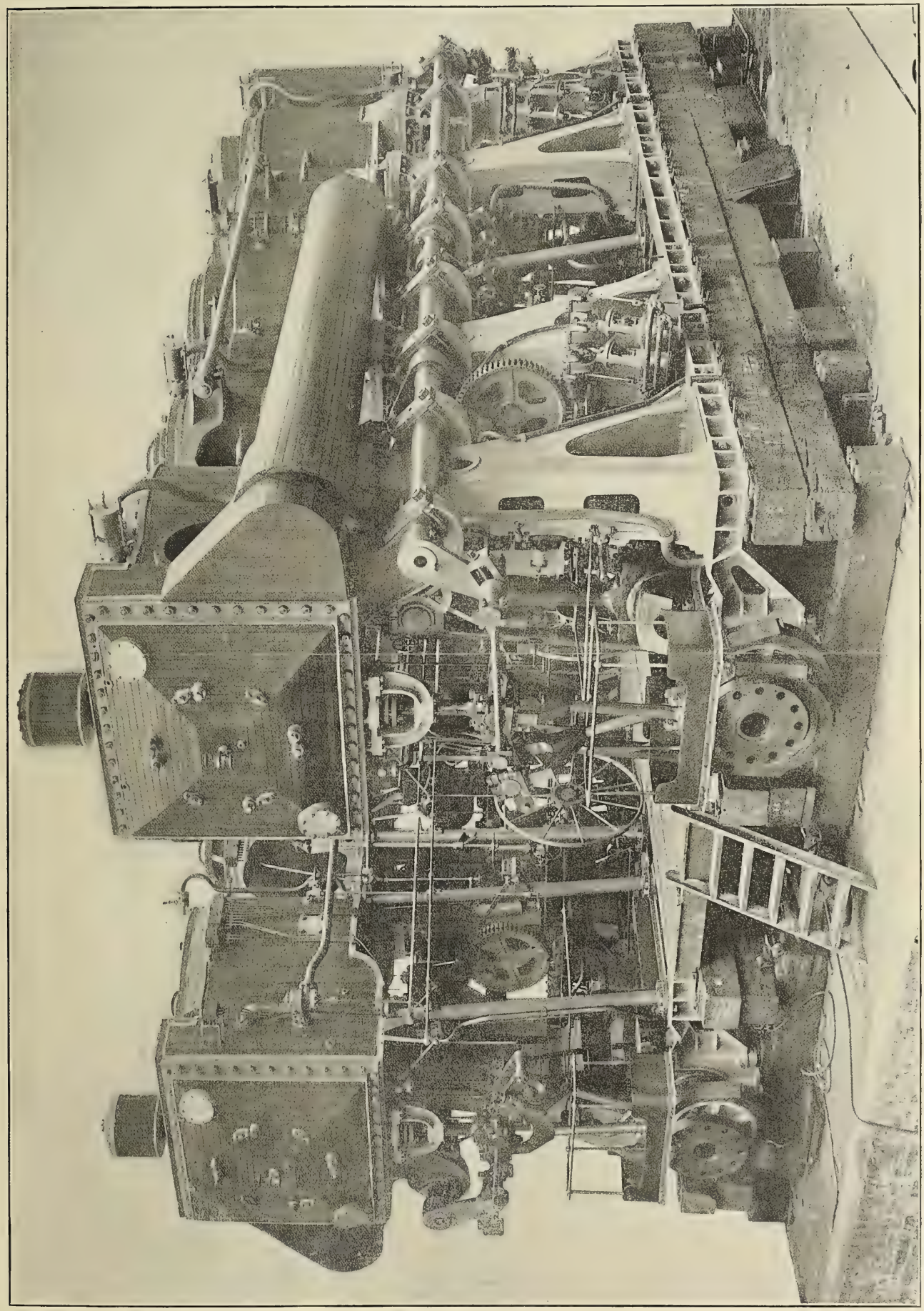
We reproduce from *Engineering* a photograph of the twin engines of one of the latest armored cruisers in the British navy, which is now undergoing completion. The ship has a displacement of 10,700 tons at a mean draft of 24 feet 9 inches. She measures 450 feet in length between perpendiculars, 68 feet 6 inches in beam, and has a molded depth of 38 feet 6 inches. The engines are designed to develop a maximum of 21,000 horsepower, at which power the designed speed is 22.25 knots.

There are two sets of four-cylinder, triple-expansion engines, arranged as usual in separate water-tight compartments. The diameters of the cylinders are as follows: high-pressure, 41 1-2 inch; intermediate-pressure, 65 1-2 inch; and low-pressure, two of 73 1-2 inch; all having a common stroke of 42 inches. All of the cylinders are steam jacketed and fitted with liners. The pressure carried is 205 pounds at the engines, and a speed of revolution of 138 per minute will be required to give the designed speed of the ship.

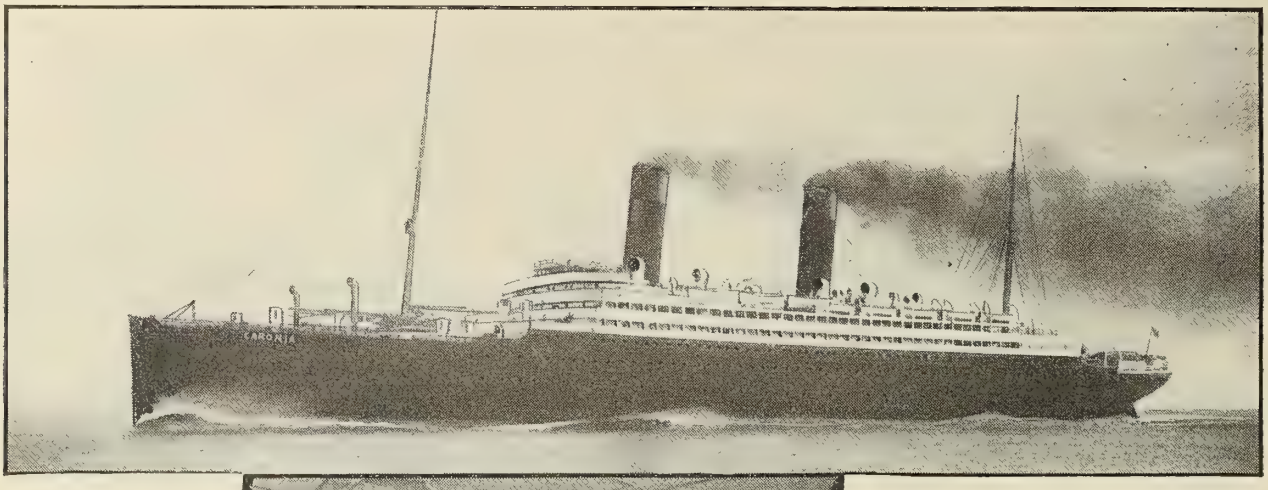
The high-pressure and intermediate-pressure cylinders are fitted with piston valves, and the low-pressure cylinders with flat side valves; the former are 28 1-2 and 46 1-2 inches respectively in diameter. The main steam pipe to each engine has a diameter of 16 1-4 inches and the exhaust of each low-pressure cylinder has a diameter of 28 inches. The high-pressure and intermediate-pressure piston rods are 9 1-2 inches in diameter, while the low-pressure piston rods have a diameter of 7 1-2 inches.

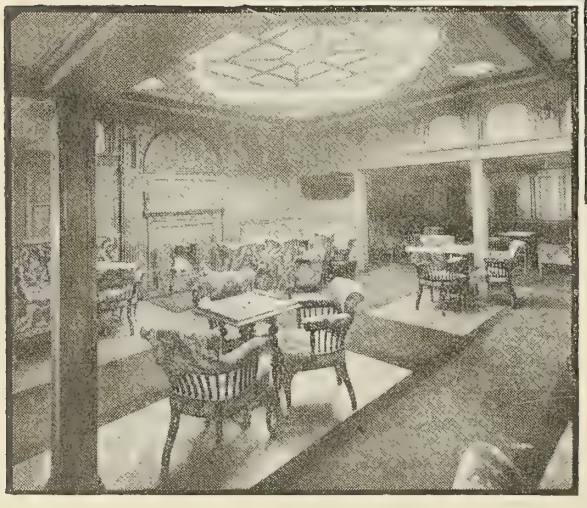
The cylinders are supported at the front by eight forged steel columns, and at the back by four cast-iron columns formed with guide faces, and one forged-steel column. The forged-steel columns have diameters of 6 1-2 and 5 3-4 inches. The bed plate is made of cast steel of I-section, and has in it eight main bearings of an aggregate length of 14 feet 8 inches. The shafting is hollow throughout, the crank shaft being 18 inches in diameter with a 9-inch hole. The crank pins have a diameter of 20 inches and a 10-inch hole, those for the high-pressure and intermediate-pressure cylinders being 26 inches long, and for the low-pressure 17 inches. The thrust shaft is 17 inches in diameter with a 9 1-2-inch hole.

Steam is provided by sixteen Babcock and Wilcox water-tube boilers with an aggregate heating surface of 45,847 square feet, and a total grate surface of 1,177.5 square feet, the ratio of heating surface to grate area being thus 39 to 1.



THE TWIN ENGINES OF THE ENGLISH ARMORED CRUISER ARGYLL.





THE CARONIA.

FIRST CABIN SALOON.

SMOKING ROOM.

THE NEW CUNARD STEAMSHIP CARONIA.

Leaving Liverpool on the 25th of February, the latest of the Cunard steamships reached this port, on her maiden trip, on the 5th of March. The *Caronia* is considerably the largest of the Cunard liners now in service, having a length over all of 678 feet, a beam of 72 feet 6 inches, a depth to shelter-deck of 52 feet, to boat-deck 80 feet, and to bridge-deck 97 feet. Her gross tonnage is 21,000, which is almost identical with that of the *Cedric* and *Celtic* of the White Star Line, and the *Minnesota* and *Dakota* of the Great Northern Steamship Line, and but little short of the 23,500 of the *Baltic,* which is the largest ship afloat. On a draft of 33 feet the *Caronia* displaces 31,200 tons.

She is one of two sister ships, the other being the *Carmania* launched on the 21st of February, and to be propelled by turbines. The keel plates were laid in September, 1903. The *Caronia* was launched the following April, and thus has started on her life work only seventeen months after the laying of the keel. Her launching weight was 13,500 tons, while the completed ship, exclusive of coal and cargo, weighs 20,000 tons. She is built with a cellular double bottom, which is carried well up the sides of the ship above the bilges, and which enables her to carry 3,500 tons of water ballast, and thus make up during a voyage for the consumption of coal and stores and maintain the steadiness with which she started out. For 250 feet amidships the hull is fitted with bilge keels, and the combination of structural stiffness, bilge keels, and provision for water ballast will doubtless result in a remarkably staunch and seaworthy ship.

The side plating is put on in strakes varying from 60 to 66 inches in breadth and 3-4 to 1 1-8 inches in thickness, and with a maximum length of plate of 32 feet, the average weight of plates being about 3 tons. The rivets in the hull are all of mild steel and number about 1,800,000; some of those used in the bottom, where there are doubling strakes along the keel, are as much as 7 inches in length and weigh each 3 1-2 pounds. The vessel is fitted throughout with the Stone-Lloyd system of water-tight bulkhead doors, by which all the passages through the numerous thwartship bulkheads may, if necessary, be closed simultaneously from the bridge, and by means of which each separate door may be opened or closed at will. An automatic device in each compartment operates to close the door of that compartment whenever a certain depth of water is reached in it. By this method the most complete security obtainable is insured. In order to counteract the effects of corrosion within the skin of the ship, particularly in the ballast tanks, bunkers, and other remote portions of the hull, the hold has been coated with bitumastic enamel.

The *Caronia* has eight continuous decks, the uppermost being the boat-deck, the others, leading downwards, the upper promenade, promenade, saloon, upper, main, lower, and orlop decks. For the convenience of passengers not acquainted with the nautical names given these decks, each one has also received a letter, beginning from above, A, B, C, D, etc.

The navigating bridge, which surmounts the main superstructure carrying the first and second-class quarters, is built in and faced with glass, so as to afford protection to those in charge of the ship, and to secure better results by promoting the comfort of those responsible for her direction. A large number of telegraphs and telephones are located in the bridge house, in order to facilitate communication with other parts of the ship as it may be necessary. Beneath the navigating bridge is a steel house on the boat-deck, the forward end of which is fitted up for the

captain and chief officer, and the after end for the junior officers. These all connect with both the outside-deck and the saloons below. A Marconi station is situated just forward of the after funnel, on the boat-deck.

The forward end of the boat-deck, as well as the two decks next below, are set aside as a promenade for the saloon passengers. Connecting with the boat-deck, somewhat aft of the officers' quarters and immediately aft of the forward funnel, is an elaborately fitted and furnished apartment, paneled in mahogany and old-gold leather, the upholstery and carpeting being uniformly of a dark shade of peacock-blue green. This apartment, which measures about 36 feet square, is known as the "lounge," and forms a common meeting point for passengers, in the most informal way. At the after end of the lounge is a buffet whence coffee and light refreshments may be obtained.

Below this are found a number of first-class state rooms of the highest grade, which are, generally speaking, unsurpassed by anything afloat both in roominess, decoration, furnishings, and

The promenade-deck contains a great many first-class state rooms, as well as the barber shop, doctor's room, dispensary, typewriter room with stenographer, and a new institution—a candy store. The purser's department is a commodious office on this deck, and is fitted up very much like the counting room in a modern bank. A steel-lined treasure chamber, and an immense safe, are provided for the custody of such valuables as the passengers do not care to leave in their state rooms.

The first-class dining saloon is a very spacious apartment located amidships on the saloon or "C" deck. It extends from side to side of the ship, and for a length of nearly 70 feet. The arrangement of the tables is somewhat different from that usually in vogue, there being a circular table in the center and numerous smaller tables arranged around it, the whole accommodating the entire number (300) of first-class passengers carried. The wing tables are in alcoves, facilitating the gathering of small parties with considerable privacy. The decoration of this room is very largely in carved mahogany, with white enamel finish, and the

THE CARONIA ON TRIAL TRIP.

appointments. Many of these rooms are connected *en suite,* comprising sleeping and sitting rooms, bath, and lavatories.

On this deck (upper promenade, or "A,") are also the writing and drawing rooms, in both of which the wainscotting is white with tapestry paneling. The ceilings are of the same color, and the rooms are plentifully supplied with luxurious couches, settees, and easy chairs, the general effect of comfort being much enhanced by the presence of an open fireplace. The carpeting, of rich red, is in perfect harmony with the curtains, the drapery at the two entrances, the table covers, and the shades of the lamps. These two rooms open into each other, thus forming virtually one apartment, measuring about 47 feet forward and aft and 32 wide. The situation is amidships, between the two funnels.

At the after end of this deck is situated the smoking room, surmounted by a fine skylight. The paneling and furnishing is in dark oak and the upholstery in figured cloth of mellow colors. A large, open fireplace, with mantel of hammered copper, adds a home-like touch to the appointments. The dimensions are about 30 by 48 feet, the length being athwartship.

furniture is very rich. The roof of this room is broken by a large octagonal opening, 24 feet across, forming a "clerestory," and surrounded above by a lounge. The angles of this opening are supported by fluted Ionic pillars, interspersed with panels in Jacobean carving. The same decorative scheme, in somewhat higher relief, is shown in the ceiling and in the frieze which surrounds the entire room, and harmonizes perfectly with the capitals of the columns. The draperies and upholstering are of a rich crimson, adding a decided warmth to the general tone.

The accommodation provided for the second-class passengers has been worked out on an elaborate scale, and has, indeed, been considered by some to be so far advanced as to be likely to cut down the receipts from first-class passengers. The third-class travelers are berthed in two-, four-, and six-berth rooms. The ship has a capacity for carrying nearly 3,100 passengers, together with a crew of 450 men.

The ventilation of the vessel is a strong feature, and has been worked out with the idea of securing natural ventilation wherever possible. In such parts of the ship as cannot be reached by this

method, the most up-to-date apparatus is employed, and splendid results are anticipated. The heating of the ship has also received much attention.

The engines of the *Caronia* are of the quadruple-expansion reciprocating pattern, consisting of two sets, each capable of generating about 10,500 horsepower. The cylinders measure respectively 39 inches, 54 1-2 inches, 77 inches, and 110 inches in diameter, with a common stroke of 66 inches. The rotations of the shaft are 86 per minute when the engines are developing their full power. There is very little of the unusual in the engine rooms, but the set of auxiliary machinery is very complete, and the laying-out of the various items making up the total has been carefully done. The condensers are separated from the engines, and raised much higher than is usually the case, so that the view from the starting platform into the wings is much more open than is usual. The engines are balanced on the Yarrow-Schlick-Tweedy system with a view to minimizing as much as possible the vibration due to operation.

There are six stokeholds especially fitted with a system of ventilation partly natural and partly artificial, in order to produce much cooler quarters than are usually found in this department of transatlantic liners. The boilers number altogether thirteen, eight double-ended and five single-ended, and contain fifty-four furnaces, supplying steam to the engines at a working pressure of 210 pounds per square inch; a slight forced draft is used when the ship is under way. The equipment of the stokehold includes the fitting of Crompton atmospheric ash hoists of 8 inches in diameter, which lift the ashes 41 feet from the stokehold floor and discharge them overboard.

The illumination is, as usual, by electricity, and is very complete, there being over 3,000 lamps, besides the powerful searchlight. The dynamos are four in number, and each is capable of generating 750 amperes of current at a pressure of 100 volts. The refrigerating machinery and cold-storage plant is one of the largest ever fitted on board ship. The engines are situated on the main-deck near the foremast, and, in addition to cooling the specially insulated holds fitted for fresh meat and other vegetables, serve also for the cooling of other compartments of the ship as may be necessary. The steering gear consists of two sets operated separately by telemeter from the bridge, either of which can be used at pleasure. One of these engines is in a steel wheelhouse on the shelter-deck aft, while the other is situated below the water-line, in order to meet the requirements of the Admiralty. Both are driven by steam power, and when one is in operation the other is automatically disconnected, and thus in no way interferes with the free action of the rudder. The upper gear is fitted over the rudder head, the lower gear being attached to a crosshead fitted on the rudder head. The latter is 19 inches in diameter, and the rudder, which is balanced, is of the most massive character, and is entirely below the water-line. The engines which work the rudder are carried on the tiller, and operate a pinion wheel acting directly on a massive toothed quadrant. Great power, steadiness, and strength are secured by this arrangement. The ship can be steered from either the wheel-house on the bridge, the shelter-deck, or the lower deck, the steering orders in the latter case being transmitted from the bridge by telephone or telegraph.

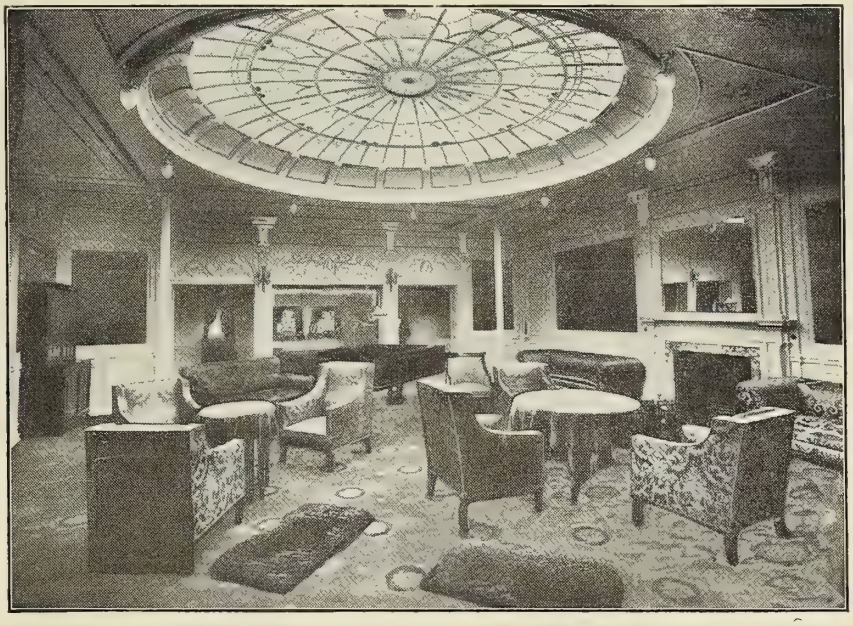
The bower anchors, which are of the stockless type, weigh each 8 1-2 tons, and have cables 3 3-8 inches in diameter. The windlass is placed forward below the shelter-deck, but the cable is carried up to the upper-deck before descending to the chain locker. Cargo stowage and unloading is made easy by the use of sixteen steam winches conveniently situated near the hatchways. Facilities are provided for dealing with 13,600 tons of dead weight. The bitts and mooring posts, with which the decks of the ship are elaborately provided, are on the same massive scale as all the rest of the equipment, and appear to be most satisfactory with regard to strength and reliability.

THE LOUNGE.

The trial trip of the *Caronia* occurred on the 4th of February. The speed condition required was 18 knots on the Atlantic passage with a normal load, and it was considered sufficient to stipulate that 19 knots should be realized on the measured mile, and maintained subsequently on a long-distance sea run. The mean speed on four runs on the measured mile was 19.62 knots, attained with the engines making an average of 89.2 revolutions and indicating 21,870 horsepower. Following these runs, the vessel proceeded to sea, and the same power and revolutions were continued for 13 1-2 hours, when the officials of the Cunard Line expressed themselves satisfied. Indicator cards were taken at regular intervals during the trial, and the highest power recorded was 23,500 horsepower. The results of the progressive speed trials are also now available. The speed of 15.72 knots was realized with one-

half the engine power—10,400 horsepower—so that the four extra knots involved a doubling of the power. For 16.61 knots the power was 12,350 horsepower, for 17.76 knots it was 15,500, for 18.80 knots 18,900, and on this series the full speed of 19.51 knots required 21,600 horsepower. The amount of vibration recorded proved that it had been reduced to the minimum by the perfect balancing of the engines. These will be most instructive when compared with similar data obtained from the turbine-driven sister ship *Carmania*.

The *Caronia* has been specially designed for utilization by the British Admiralty as an armed cruiser, provision being made for the installation of twelve 6-inch rapid-firing guns whenever any emergency requires that the ship be withdrawn from her regular service and sent out with the fleet.



THE FIRST-CLASS DRAWING ROOM.

MARINE ENGINEERING DATA CARD. APRIL, 1905.

NAME—Caronia. NATIONALITY—British. TYPE—Ocean passenger and freight liner.
LAUNCHED—1904. BUILDER—John Brown and Co., Ltd. OWNER—Cunard Steamship Co., Ltd.

Length over all	678 ft. 0 ins.
Beam, molded	72 " 6 "
Depth, " to shelter deck	52 " 0 "
Launching, displacement	13 500 tons
Load, draft	33 ft.
" displacement	31,200 tons
Dead weight cargo capacity	12,000 "
Gross tonnage	21,000 "
Water ballast	3,400 tons
Speed at sea, designed	18 knots
" on trial	19.62 "

Engines: Type	4-cylinder, quadruple
Cylinder diams	39, 54 1-2, 77, 110 ins.
Stroke	66 "
I. H. P., designed	21,000
" on trial	21,870
Shafting, diameter crank	22 1-2 ins.
" " tunnel	21 1-4 "
" " propeller	23 1-2 "
Boilers: Type	Scotch
Number	8 double end; 5 single-end
Steam pressure	210 lbs.
Draft	Howden system
Stacks above grate	135 ft.

A Collision on the Clyde.

The steamer *Kathleen*, of 1,017 tons registry, bound from Bilbao, Spain, to Glasgow, with a cargo of iron ore, and almost at her destination, was struck amidships by the steamer *Stromboli*, of 727 tons dead weight, bound for Valencia with a cargo of machinery, and almost cut in two. As a result both ships sank, but not before the *Stromboli* had succeeded in pulling the two wrecks into shallow water near the shore. We present a photograph showing the condition of the two ships at low water, at which

time it was ascertained that the ships were "toggled" together, the side plating of the *Kathleen* having pierced the bow of the *Stromboli* while the fore foot of the latter had punctured the engine bulkhead of the former. It is thought that the *Stromboli* may be saved, but as the other ship was cut almost entirely in two, she will probably prove a total loss. The photograph was taken by Mr. I. S. Willet of Greenock.

PRACTICAL POINTS ABOUT THE SCREW PROPELLER.

BY W. F. DURAND.

PART IV. *General Discussion of Relation Between Proportions of Propeller, Conditions of Operation, and Results to be Expected.*

(1) *Thickness of Propeller Blades.*

The thickness of the blade at the root has already been used in Part II, in connection with the laying down of the propeller on the drawing board. To determine its dimensions we may use the following formula:

$$t = A\sqrt{\frac{B}{n}}\,du.$$

Where t = thickness of blade at root in *inches*,
A = coefficient according to material, as given below,
B = coefficient drawn from table below,
n = number of blades,
d = diameter of propeller in *feet*,
u = speed in knots.

VALUES OF A.

For bronze or steel	.04 to .06
For cast iron	.07 to .08

VALUES OF B.

Pitch-ratio.	B.	*Pitch-ratio.*	B.
1.0	.91	1.6	.56
1.1	.83	1.7	.53
1.2	.76	1.8	.50
1.3	.69	1.9	.47
1.4	.64	2.0	.45
1.5	.60		

Thus for example:

Let diameter = 10 feet, u = 15 knots,
n = 4, pitch-ratio = 1.2,
material, cast iron.

Then $t = .075\sqrt{\frac{.76}{4}} \times 10 \times 15.$

$= .075 \times .44 \times 10 \times 15 = 5.0$ inches.

Again, if diameter = 8 feet, u = 28 knots,
n = 3, pitch-ratio = 1.7,
material, bronze.

Then $t = .045\sqrt{\frac{.56}{3}} \times 8 \times 28$

$= .045 \times .43 \times 8 \times 28 = 4.33$ inches.

(2.) *General Influence of the Four Factors a, b, c, e.*

These four factors, as discussed in the preceding chapter, have reference respectively to the influence of the ship-speed relation; the pitch-ratio; the slip; and the shape-area, or general form and proportion of the blade.

As the Admiralty coefficient is smaller and the ship-speed factor is larger, the propeller must be correspondingly larger, and this means, with all other considerations equal, that the propeller must be the larger, as the speed is the more disproportionate to the ship either in form or size. Thus a full-bodied small ship, if pushed to a relatively high speed, will require a relatively large propeller, and no juggling otherwise can fully compensate for a mistake made at this point.

Again, all other conditions remaining the same, it will be seen that the larger the pitch-ratio the larger the resulting diameter, and *vice versa*. In a general way, then, high pitch-ratios will carry with them large diameters, and are therefore more directly indicated when, due to other general conditions, such diameters are acceptable. If the diameter is to be reduced, then the reduction of the pitch-ratio and the acceptance of higher revolutions is one of the means available for bringing about this result.

The slip factor, on the other hand, varies inversely as the slip, which means that the larger the slip, the smaller the propeller, and *vice versa*. Working at a low slip always means a large propeller, and with a large slip a relatively small propeller.

Likewise, the more the area, with certain limitations as previously discussed, the smaller the resulting propeller, and *vice versa*. Thus, likewise, in a general way, a propeller with three blades will be of greater diameter than one of four, and one of two larger than one of three, simply because the less the number of blades the less the area readily carried and the greater the resulting diameter.

To sum up, it will be seen that with a given ship and set of general conditions, the smallest possible propeller will result with a low pitch-ratio operated at high slip and with a relatively large surface. This would not mean presumably an efficient propeller. It would simply represent the combination of all items tending toward small size, and at the same time relatively high revolutions. On the other hand, the opposite combination of high pitch-ratio, low slip and small surface would give an excessively large propeller and relatively low revolutions. Neither would such a combination necessarily give a propeller of good efficiency.

We will now turn briefly from size to efficiency in its relation to the various items under discussion.

It should be first understood that efficiency is chiefly a question of slip in relation to pitch-ratio. In a general way the best efficiency may be looked for with values of the apparent slip from 14 to 20 percent; the higher the pitch-ratio, the higher the permissible slip for good efficiency. In particular it should be understood that decreasing the slip will not necessarily increase the efficiency, and certainly not beyond the value for the best efficiency. Still lower values of the slip will give decreased efficiency and greater operative loss throughout. Too low slip is therefore quite as undesirable as too high, and, in fact, experiment shows that efficiency falls off more rapidly as the slip is reduced below its best value than when it is increased above such best value. It will be therefore better to operate at five percent above the best value of the slip than at five percent below. Variations of area within moderation have but slight influence on the efficiency, and area may be added freely up to values of the area-ratio as large as .50 without fear of ill effect on the efficiency, except for low pitch-ratios. Here, in a general way, large areas should be avoided, and for values of the pitch-ratio about unity, the maximum area-ratio should not much exceed .40. From these considerations we may readily derive certain indications regarding the steps to be taken in case the trial results are different from those expected or required. Thus suppose: (*a*) The boilers furnish plenty of steam, but the revolutions are too low, the expected indicated horsepower is not developed, and the speed is not attained. This condition with large slip may indicate either the engine is adequate for the ship, but pitch-ratio too high, or, on the other hand, engine is adequate for propeller, but both too small for ship. In the former case the remedy will be a decrease in pitch-ratio, and in the latter an increase in engine power and in size of propeller.

This same condition with small slip may imply engine adequate for ship, but propeller is of too large diameter, or too much surface, or both.

With the slip lying within the normal working range, this condition may indicate engine adequate for ship, with some combination of too high pitch-ratio, too large diameter and too much area; or, on the other hand, propeller adequate for ship but engine too small. Usually the special conditions will make it possible to discriminate between these double cases.

In all these cases it must be understood that the engine cannot develop the power unless the propeller will allow it to make the revolutions, and any condition which tends to hold the engine down below its designed revolutions will by this very fact tend to prevent the development of the indicated horsepower, and hence the realization of the speed.

(*b*) The revolutions rise above the designed figure, running down the steam pressure, and working off all the steam the boiler will furnish, but without giving the expected speed. In such case nearly the full engine power is developed, but the excessive revolutions will presumably mean a corresponding increase in the slip and a falling off in the efficiency. This will result in a waste of power, and in failure to realize the full speed. The remedy will be a larger propeller, or more surface, or both, in order to hold the revolutions and slip down to values which will give the propeller a chance to apply with good efficiency the power which it receives from the engine.

(*c*) The expected revolutions are obtained, but the speed is low and the slip excessive. This shows that the propeller is absorbing all the power that the engine is developing, but that the efficiency is presumably low, and the total power probably too small for the purpose. Such a result might, for example, be obtained on a large ship with an engine and propeller entirely too small. The remedy means more power at the start and a larger propeller for its economical application. If, however, no more power can be supplied, and it becomes a question of doing the best possible with the power at hand, then some gain can be realized by applying such power as is available with the best possible efficiency, and accepting such reduced speed result. This means a reduction of the slip, and this, in turn, means some combination of the following items: larger diameter, lower pitch-ratio, more surface and perhaps higher revolutions.

The same general relations may also be stated with reference to the results of poor design. First assuming the engine adequate for the ship we have as follows:

(I.) With suitable diameter and pitch-ratio and too little surface, the revolutions and slip will be high, especially at the upper revolutions, and the speed will fall short.

(II.) With suitable diameter and pitch-ratio, but too much surface, the revolutions may fall below the proper value and the engine may fail to develop the full power, and the speed will therefore fall short.

(III.) With suitable diameter and surface, but too high a pitch-ratio, the revolutions will be too low, the engine will not develop the power, and the speed will fall short.

(IV.) With suitable diameter and surface, but too low a pitch-ratio, the revolutions will overrun; the steam pressure will be run down, the engine will not develop the power efficiently, and it may be possible that the slip will fall below the value for the best efficiency; or, on the other hand, it may not be permissible to run the engine at the higher speed. In either case the speed of the ship may suffer in consequence.

(VI.) With suitable pitch-ratio and surface, but too large diameter, the revolutions will overrun with increasing slip, and the steam pressure will be run down, both of which will react unfavorably on the power finally applied on the ship. On the other hand, if the higher revolutions cannot be allowed, then the engine must be held back, the full power will not be developed, and the speed will fall short.

(VI.) With suitable pitch-ratios and surface, but too large diameter, the revolutions will fall short, full power will not be developed, the slip may fall below the value for best results, and the speed will fall short.

(VII.) If, on the other hand, we have the engine adequate for the propeller, but both too small for the ship, the result will be low revolutions and speed and high slip.

(VIII.) Again, if the propeller and ship are properly matched but the engine power is too low, the result will be low revolutions and speed and normal slip.

In actual cases the result will often depend on some combination of the several items. A careful study of these various cases from both points of view as given above should, however, serve to indicate the probable nature of the trouble in any given case, and thus point the way to the changes required for improvement.

(3.) *Number of Blades.*

So far as experience may determine, there seems to be no reason for assuming that the number of blades, in itself, is an item of great importance. Indirectly, the number of blades affects the result by its influence on the amount of surface which can be comfortably carried. There is every reason to believe that between two propellers of the same surface, one of three and one of four blades, all other things being equal, the difference in the performance would hardly be of practical importance. It results actually, however, that the average propeller of four blades will have more area than that of three, and the latter more than that of two. Hence, the use of two, three or four blades is partly a matter of the allowable diameter.

Where the utmost must be realized on a given diameter, four blades should be employed. Where, on the other hand, the available depth will permit a diameter suited to the area readily carried by three blades, the three-bladed propeller will usually be preferred on account of the decreased cost of construction as compared with that of four blades, especially with removable blades. The two-bladed propeller, in modern practice, is confined to the reversible types, such as are fitted with gas engines and runabout boats, launches and small yachts. With twin or multiple screws the draft usually admits readily of the three-bladed form, and such are commonly employed in such cases.

(4.) *Shape of the Blade.*

Prevailing practice based on the sound interpretation of experience justifies the use of a generally oval, or possibly pear-shaped developed form of blade. The usual types are oval, and often elliptical, or nearly so, as discussed in Part II. There seems to be no objection, however, to filling out the blade somewhat toward the tip, especially if it is left well rounded, thus giving it a form slightly tending toward the pear shape, with broad end at the tip. In particular it is desirable to avoid sharp angles and corners at the tips, as such are sure to produce eddies in the water with an accompanying loss of efficiency.

(5.) *Amount of Surface.*

The amount of surface has been already referred to in the chapter on Design. Supplementing the brief reference there made, it will be sufficient to note that in general, variation of area affects efficiency very slowly; that increase of area brings with it increase of thrust and greater general effect, but at a slower and slower rate as the area is added, until a point is ultimately reached beyond which increase of area gives no addition of thrust, and may, indeed, result in a loss of thrust and general effect; that this condition is reached at lower and lower limits the lower the pitch-ratio; that this limit will be found somewhere about .55 or .60 area-ratio for high pitch-ratios, and not far from .40 for fairly low pitch-ratios, with corresponding intermediate limits for intermediate values of the pitch-ratio; that while this limit will vary somewhat with the shape of blade and thickness, these general limits may be accepted as giving substantially all the thrust likely to be obtained without danger of loss in efficiency.

(6.) *Specially Low Pitch-Ratios.*

There has recently arisen special interest in the use of very low pitch-ratios for gasoline motor boats and in connection with the steam turbine. Such relatively small experimental data as is available seems to indicate that such pitch-ratios may be employed with fair efficiency, provided too much surface is not employed. With excessive surface the efficiency is sure to be very low indeed, with resultant loss of useful effect. The upper limit of surface area for propellers of pitch-ratio less than unity should presumably be not much over .35.

(7.) *Immersion of the Propeller.*

With insufficient immersion of the blades, and in particular if the blades cut the surface of the water, air will be drawn down

and mixed with the water operated on so that the actual material handled by the propeller will be not water, but an emulsion or mixture of air and water. This of course weighs less per cubic foot than water alone, and in consequence the thrust gained by the propeller when operating at a given slip will be decreased, with corresponding loss in effect. In general, therefore, the immersion of air. The necessary depth of immersion will increase with the of the blades should be sufficient to insure against such indraught slip at which the propeller works. For ordinary conditions the tip of the blade in its highest position should not come nearer the surface of the water than about ¾ the radius, though with moderate values of the slip smaller immersions may be safely employed. It should be noted that with the proper design in advance, the blades may be allowed to cut the surface of the water without sacrifice to propeller. It will simply call for a larger propeller—sufficiently large so that, working on the mixture of air and water, it may nevertheless obtain the needed thrust at a moderate value of the slip. Such propellers have been sometimes employed in connection with canal-boat propulsion.

(8.) *Location of Propeller Relative to Ship.*

In a general way, the propeller should be kept well clear of the stern post in order to avoid the sharp blow which may be communicated from the blade to the ship when the clearance is small, especially at high speeds. Beyond this, the farther the propeller is away from the ship, the less will be the effect of its "suction" on the ship, producing the virtual increase of resistance. On the other hand, if placed too far away, some effect may be lost by way of the wake in which the propeller works, and which returns through the propeller some part of the energy which has been spent in producing it by the motion of the ship through the water. Having in view structural and general considerations, it will usually be found desirable to set the propeller as near the ship as may be permitted by the considerations of clearance just noted.

(9.) *Number of Propellers.*

This question is one turning chiefly on the desired subdivision of power, the available draft and the desired number of revolutions of the engines. There seems to be no reason for believing that in themselves two or three or more propellers are more or less efficient than one. In each case, however, the propeller must be adapted to the conditions under which it is to work. In general, the greater the number of propellers and the less the power of each one, the smaller the propeller and shaft, the less the draft in which it can work, and the higher the revolutions for a given speed.

(10.) *Cavitation.*

This is the name which has been given to a condition which has arisen in connection with very high speeds. It is found that, beyond a certain speed and under certain conditions, the water does not close in behind the propeller blades, and that, in consequence, they work, to a greater or less extent, in the cavity thus formed within the water. This results in a decreased thrust and general loss of effect. It has been found that by a suitable provision of blade surface in proportion to the thrust this condition may be avoided. The necessary surface may be determined closely by the following equation:

$$A = \frac{D^{\frac{2}{3}} u^2}{8}$$

where A = projected area of all blades in square inches,
D = displacement in tons, u = speed in knots.

It will be noted that this is the projected area and not the actual or developed, and that it must be determined by laying the propeller blade down on the drawing board and measuring the resulting projected area.

Thus for illustration, if $D = 216$ and $u = 25$, then projected area not less than $\frac{216^{\frac{2}{3}} \times 25^2}{8} = \frac{36 \times 625}{8} = 2813$ square inches, should be employed.

(11.) *Dock Trials.*

It is a fact not always remembered that with the vessel tied to a dock the engine cannot develop the full power. This is because the propeller will hold the revolutions down and prevent the realization of the designed engine speed, and hence the development of the designed power. The explanation of this effect on the revolutions is found in the fact that with the vessel tied to the dock the slip is 100 percent, and hence the full thrust and full turning moment are reached at a lower value of the revolutions than when the vessel is in free route, and the propeller is operating at the normal slip. Now the revolutions at which an engine will run are determined by the condition when the resistance to turning the propeller will just balance the turning moment of the engine. In other words, the engine will run up to the speed at which the resistance to turning the propeller will just balance the turning moment furnished by the engine. The latter, with any given engine, is primarily dependent on the steam pressure, and hence with fixed pressure the moment remains fixed. Now there is a close and practically fixed relation between the thrust developed by the propeller and the effort required to turn it, so that, with a fixed maximum turning moment, we shall have very closely a fixed maximum thrust. Both of these furthermore depend on revolutions and slip; the higher the revolutions the lower the slip at which a given thrust and turning moment will be developed, while, *vice versa,* the higher the slip the lower the revolutions at which a given thrust and turning moment will be developed. It immediately follows, then, that if a propeller is operating under excessive slip, the maximum thrust and the turning resistance which will balance the engine moment will be reached at lower revolutions than with normal slip, and when this condition is reached, the revolutions can, of course, go no higher without a further increase in steam pressure and in the moment developed by the engine. Due to these facts, no engine can develop full power under dock trial, and the results of such trial must, therefore, not be taken as representing the power capacity of the engine.

(12.) *Power Developed when Towing.*

For similar reasons, when one ship is towing another the full power cannot be developed. The propeller, when towing, will necessarily work at a higher slip than when the ship is running free, and hence will reach the full thrust and full turning moment at lower revolutions, and hence at lower power, even with full steam pressure. See also (2) Case VII.

(13.) *Action of the Propeller in Stopping and Starting.*

The principles stated with reference to the action of the propeller in towing or in dock trials will serve to make clear its action in stopping and starting. In the latter operation, suppose the full steam pressure thrown on the engine. This means the full engine turning moment, and, in answer, the propeller will rapidly jump up to the speed at which the full thrust and turning moment will be in equilibrium with the engine, and each will remain substantially unchanged so long as the steam pressure remains the same. This condition will, furthermore, be reached at revolutions somewhat below the maximum number, and for the same reason as in a dock trial—simply because the slip is excessive and the full moment is reached with excessive slip at a lower value of the revolutions than when the slip has its normal value. The full thrust thus developed is, of course, all available for starting the ship from rest; but, as the speed gradually increases the water resistance correspondingly increases and absorbs a correspondingly larger part of the thrust, leaving a smaller and smaller part available for increasing the speed of the ship. It thus results that the gain in speed, at first large, grows less and less as the ship finally approaches her top speed, when the water resistance will absorb the full thrust, with nothing left over for gain in speed. As the ship gathers headway, moreover, the slip, at first large, will grow less and less, and the revolutions at which the full thrust and turning moment are developed will become higher and higher. The revolutions of the engine will,

therefore, increase gradually as the speed increases, coupled with a corresponding increase in power; all at a constant steam pressure, turning moment, and substantially constant thrust on the ship. In this manner the ship gathers headway, and finally reaches her top speed, at which we have the maximum revolutions and power, with the full thrust balanced against the water resistance of the ship.

Conversely, in stopping the ship the propeller develops a backward pull on the shaft, which opposes the forward motion of the ship. At the first instant of backing, therefore, the forward motion is opposed both by the water resistance and by the pull of the propeller. As the ship slows down under the combined action of these retarding forces, the water resistance becomes less and less as an opposing force, and the total retarding force approaches more and more nearly to the backward pull of the propeller. It thus results that the retardation, at first pronounced, gradually decreases until the ship is finally brought to rest under the retarding influence of the propeller alone.

(14.) *Measurement of Pitch.*

To determine the pitch of a given propeller three measurements are necessary. See figure 38. These are:

(1.) The radius OA at which the pitch is desired.

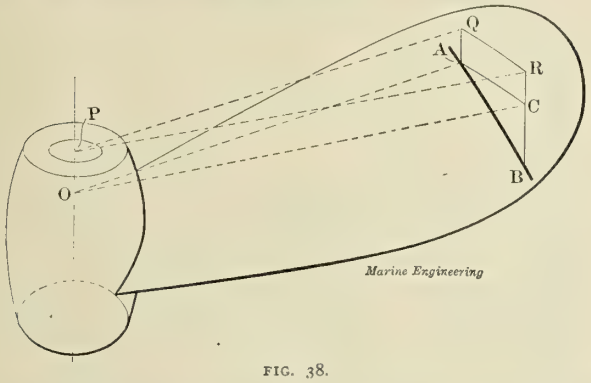


FIG. 38.

(2.) The angle or part of the complete circumference corresponding to the distance on the blade between A and B, the two points between which the pitch is to be found.

(3.) The advance BC parallel to the line of the shaft, corresponding to this part of a complete revolution.

In the figure, A and B are points on the face of the blade and are at a constant distance OA from the shaft center-line OO. AC is an arc of a circle which lies in a plane through A and perpendicular to the shaft. The angle AOC is, therefore, the one referred to in (2), and the distance BC is the corresponding advance. Then BC is the same fraction of the entire pitch that AOC is of the complete circle, or the same fraction that the arc AC is of a complete circumference with OA as radius. This complete circumference will be $6.2832 \times OA$. Hence the proportion:

$$AC : 6.2832 \times OA :: BC : \text{pitch}.$$

$$\text{or pitch} = \frac{6.2832 \times OA \times BC}{AC}.$$

It is not, however, so easy to measure AC as AB, so that we may put for AC its equal $\sqrt{AB^2 - BC^2}$ and we then have:

$$\text{pitch} = \frac{6.2832 \times OA \times BC}{\sqrt{AB^2 - BC^2}}.$$

A brief outline of the operations is as follows:

(1.) Select the points A and B at and between which the pitch is desired, making sure that they are at equal distances from the shaft center line. This can be done by squaring down from a straight edge or other reference line PQ, PR, placed across the hub at right angles with the shaft. Then measure the length AB.

(2.) The propeller being leveled up, measure the distance BC from a level through A vertically down to B. Or if the propeller cannot be leveled, measure from B in a direction parallel to the shaft, out to a line through A in a plane at right angles to the shaft. Or measure from Q down to A and from R down to B and take their difference BC.

(3.) Multiply the distance OA or its equal PQ by 6.2832, and by the length BC, all in the same units of measure.

(4.) Square the lengths AB and BC, subtract the square of the latter from that of the former, and extract the square root of the difference.

(5.) Divide the result found in (3) by that found in (4), and the quotient will be the pitch desired.

Thus, for example, suppose

$AB = 20$ inches, $BC = 13$ inches and $OA = 48$ inches.

Then $6.2832 \times 48 \times 13 = 3920.7$.

Also $\sqrt{400 - 169} = \sqrt{231} = 15.2$.

Then $3920.7 \div 15.2 = 238$ inches $= 21$ feet 6 inches $=$ pitch.

If the pitch is variable instead of uniform, the operation is precisely the same, but the result found must be considered merely as the mean or average value of the pitch between the points A and B. For other parts of the blade a similar process will give the pitch at those points.

When the propeller is in place on the ship it is sometimes more convenient to carry out the principles involved in this method of measuring pitch somewhat differently, as follows: Let the propeller be turned so as to bring one of the blades horizontal. Then select the place at which the pitch is desired, and hang over the blade at this point a cord with two weights, as shown in figure 39. Care must be taken that the two points A and B at

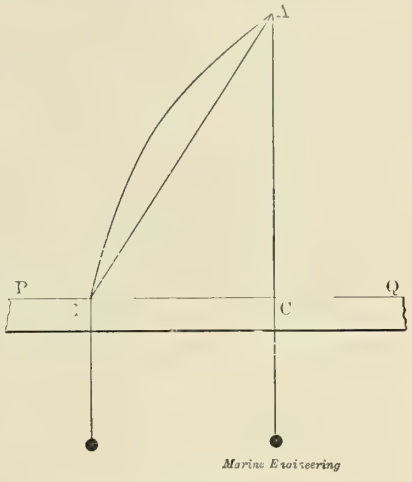


FIG. 39.

which the cord touches the edges of the blade are at the same distance from the center. It is then readily seen that the points A and B of figure 38 correspond to the similar points of figure 39, except that in figure 39 they are of necessity taken on the extreme edges of the blade. We then level up a bar PQ and measure the distances, AB and BC, as noted above, using them in the same way for finding the pitch. Or we may measure AC directly and use this with BC in the proportion above.

As a rough and ready rule, it may be remembered that the pitch of a propeller will equal the length of a circumference at the place on the blade where the slope of the face is 45°. At this point let the radius be r. Then pitch $= 2\pi r = 6.2832\ r$. In this way an approximate idea may often be quickly obtained of the pitch of a wheel by estimate without special measurement, except for the radius or diameter at which the blade has the slope of 45°.

The details of the above methods for finding pitch may vary considerably, but the description given will serve to show the principles involved, and with reasonable mechanical skill no trouble will be found in carrying out the measurements required.

(15.) *Thrust on Propeller.*

It is frequently desired to ascertain the thrust under which

a given propeller is working. To this end if the indicated horsepower and speed are known, we may use the following formula, which will be found sufficiently exact for most practical purposes:

$$T = \frac{220H}{u}.$$

Where T = thrust in pounds, H = indicated horsepower, u = speed in knots.

Thus, for example, if $H = 1200$ and $u = 15$, we have

$$T = 220 \times 1200 \div 15 = 17{,}600 \text{ pounds.}$$

If, on the other hand, we have the displacement and speed, we may use the formula:

$$T = \frac{220D^{\frac{2}{3}}u^2}{K}.$$

Where T = thrust in pounds, D = displacement in tons, u = speed in knots, and K = Admiralty coefficient as discussed in the chapter on Design in connection with the factor a.

Thus, if $D = 512$, $u = 20$, $K = 240$,

$$T = 220 \times 64 \times 400 \div 240 = 23{,}467.$$

Since, moreover, the value of K is often about 220, more or less, it will often happen that the product $D^{\frac{2}{3}}u^2$ by itself will furnish a first approximation to the value, and sufficiently exact for an immediate purpose or where only the general character of the result is desired.

Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, report under date of March 10, 1905, the following percentage of completion of vessels building for the United States Navy:

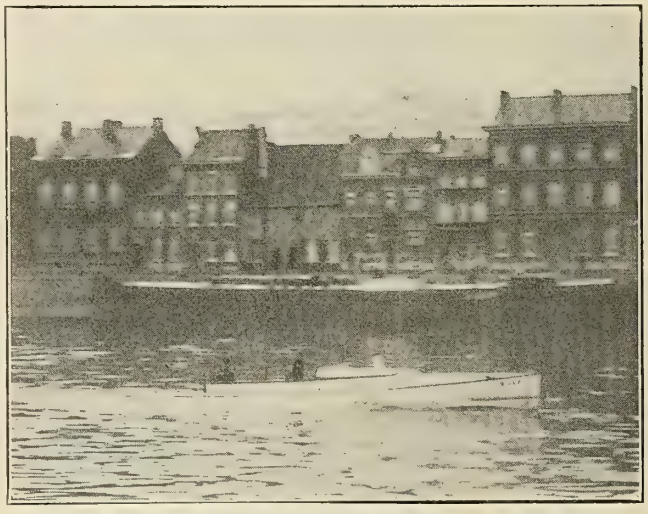
			Feb. 1.	Mar. 1.
BATTLESHIPS.				
Virginia	19 knots.	Newport News Co	76.96	80.82
Nebraska	19 "	Moran Brothers Co.	65.8	67.59
Georgia	19 "	Bath Iron Works	73.61	75.09
New Jersey	19 "	Fore River Shipbuilding Co.	75.7	77.9
Rhode Island	19 "	Fore River Shipbuilding Co.	80.7	82.5
Connecticut	18 "	Navy Yard, New York, N. Y.	64.51	67.17
Louisiana	18 "	Newport News Co.	67.8	69.54
Vermont	18 "	Fore River Shipbuilding Co.	35.4	38.8
Kansas	18 "	New York Shipbuilding Co.	41.3	43.5
Minnesota	18 "	Newport News Co.	55.7	58.69
Mississippi	17 "	Wm. Cramp and Sons	20.6	22.55
Idaho	17 "	Wm. Cramp and Sons	18.09	19.98
New Hampshire	18 "	New York Shipbuilding Co.	0.0	0.0
ARMORED CRUISERS.				
West Virginia	22 knots.	Newport News Co.	99.25	99.75
California	22 "	Union Iron Works	70.	72.2
Maryland	22 "	Newport News Co	96.5	97.5
South Dakota	22 "	Union Iron Works	67.9	69.5
Tennessee	22 "	William Cramp and Sons	63.13	65.14
Washington	22 "	New York Shipbuilding Co.	62.7	66.4
North Carolina	22 "	Newport News Co	0.0	0.53
Montana	22 "	Newport News Co	0.0	0.55
SEMI-ARMORED CRUISERS.				
St. Louis	22 knots.	Neafie and Levy Co.	58.5	58.8
Milwaukee	22 "	Union Iron Works	66.7	68.
Charleston	22 "	Newport News Co.	90.22	92.02
PROTECTED CRUISERS.				
Chattanooga	16½ kts.	Lewis Nixon	99.13	99.13
Galveston	16½ "	Wm. R. Trigg Co	96.	98.
GUNBOATS.				
Dubuque	12 knots.	Gas Engine and Power Co.	85.7	88.2
Paducah	12 "	Gas Engine and Power Co.	76.2	78.6
TRAINING SHIPS.				
Cumberland	Sails	Navy Yard, Boston	91.	94.
Intrepid	"	Navy Yard, Mare Island	78.	86.
TRAINING BRIG.				
Boxer	Sails	Navy Yard, Portsmouth	97.	98.
TORPEDO BOATS.				
Stringham	30 knots.	Harlan and Hollingsworth	99.	99.
Goldsborough	30 "	Wolff and Zwicker	99.	99.
Nicholson	26 "	Lewis Nixon	99.	99.
O'Brien	26 "	Lewis Nixon	99.	99.

THE AUTOMOBILE BOAT HILDA.

BY L. RAMAKERS.

The John Cockerill Company has just built, in its naval works at Anvers, an interesting automobile boat designed for use in the study of the application of internal combustion motors to the propulsion of rapid boats. Although this boat should be classed, on account of the power of its engine, in the category of racers, it is distinguished from these by the fact that not everything has been sacrificed to speed; the hull, very strong, is constructed of steel; the motor, also strong in every part, turns at the relatively low rate of revolution of 900 per minute. In fact the boat is not a racer, but is simply a very rapid cruiser.

The hull measures 9.45 meters (31 feet) in length; 1.20 meters (3 feet 11.2 inches) in beam, and 0.61 meters (24 inches) in depth. The shell, which is constructed of sheet steel, is supported by frames of steel angles spaced 50 centimeters (19.7 inches) apart. There are three water-tight bulkheads, which renders the boat absolutely unsinkable, even in case of the destruction of the bow in collision, for the first water-tight bulkhead, called the collision bulkhead, would limit to a very small quantity the volume of water introduced. The gasoline reservoir, made of galvanized sheet steel, is riveted and brazed. It is placed between two water-tight bulkheads, which renders a disastrous fire practically impossible.

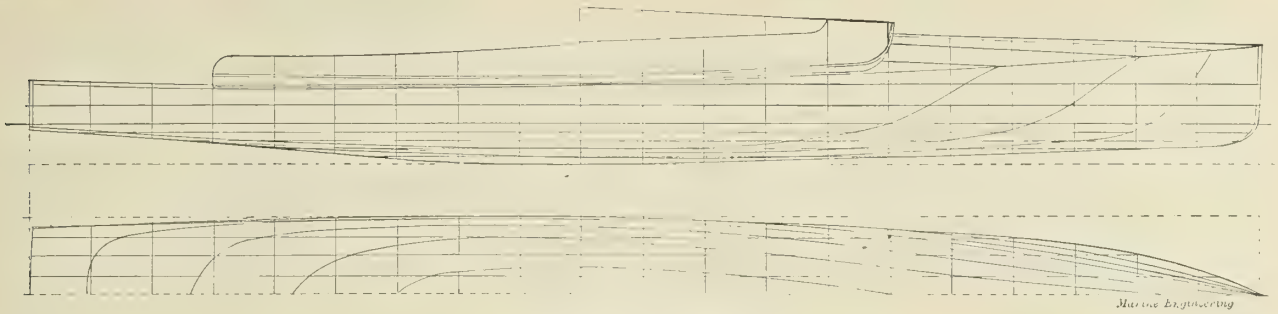


THE HILDA AT FULL SPEED.

The motors, accumulators, and all of the mechanical organs are completely sheltered under an aluminum hood, which may be dismounted in a few seconds, or put in place simply by means of two nuts of simple and rapid action. The exhaust pipe is led under the base of a large funnel which projects above the hood. The discharge of the burnt gas produces in this funnel a violent current of air, the effect of which is to continually clear the hood of the fumes produced by the motor, to restrict an increase in the temperature, and to prevent the accumulation of an explosive gas from the accidental leakage of the carburetor or from a leakage of the gasoline pipes. In fact, the gas is diluted to such an extent by air, that the odor, at worst very slight, disappears completely.

Those who, in order to cool their motors, have employed coils of pipe, will recognize the fact that this disposition of the funnel has a very practical value, the inconvenience arising from the former practice being especially the introduction with the air for combustion of a certain amount of water whenever the sea is at all rough, and, in addition, the drawing down upon the pilot of all the fumes and odors from the motor.

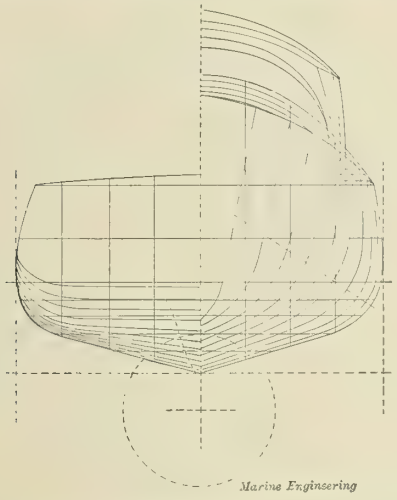
The plans which we publish show much better than could a long description the disposition of the machinery on board. The plan of the lines of the boat is of considerable interest because of the excellent results given on trial. As may be seen, the water-lines are



THE LINES OF THE HILDA.

decidedly straight forward, while the stern is completely flat, this being in contrast with the general practice of boat builders on the continent.

The engine is of the Standard type, 35 horsepower, and was built by the Germain Company at Charleroi, Belgium. It has four cylinders, with suction valves so controlled as to permit a variable rise of valve, thus allowing the motor to run constantly at a prearranged speed anywhere from 180 to 900 revolutions per minute, without having to regulate through the carburetor. The



THE BODY PLAN.

reversal of direction is made by means of a differential gear, the operation of which gives complete satisfaction. The screw is of a simple form, with a diameter of 50 centimeters (19.7 inches), and a constant pitch of 66 centimeters (26 inches). The generatrix is right-handed and inclined toward the rear at an angle of 78 degrees. The material is bronze.

The results given at the trial of this boat upon the river Meuse, at Liège, in calm weather, are as follows:

Load, three men and 68 kilograms (150 pounds) of gasoline.

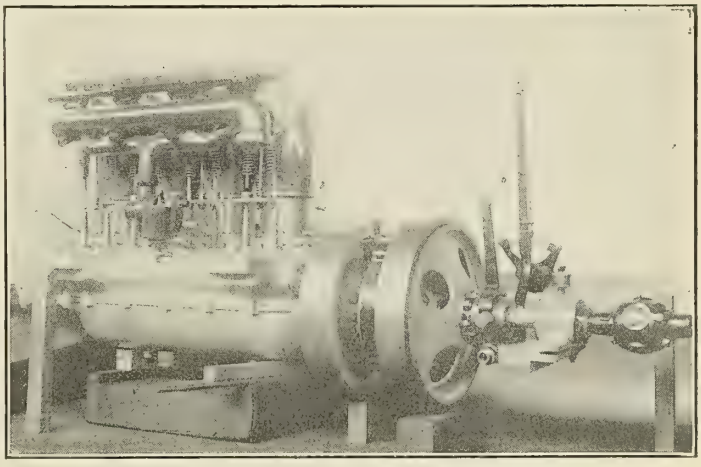
Total displacement, 1,604 kilograms (3,530 pounds).

Freeboard amidships, 27 centimeters (10.6 inches).

Horsepower of engine as given by a trial in the shops, 45, at 900 R. P. M.

Speed upon an officially measured course of 3,530 meters (3,860 yards), as shown by means of the time required to cover the course: against the current, 405 4-5 seconds, giving a speed of 31.35 kilometers (16.9 knots) per hour; with the current, 395 seconds; speed, 32.2 kilometers (17.33 knots) per hour.

Absolute mean speed, 31.77 kilometers per hour, or 17.11 knots.



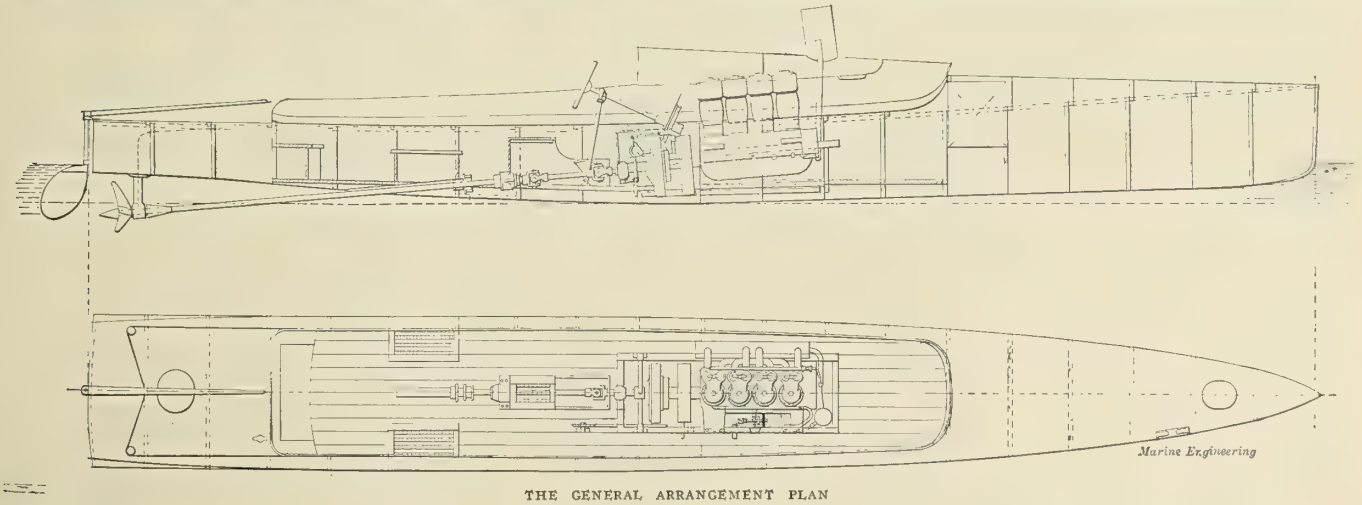
THE FOUR-CYLINDER ENGINE AND GEAR.

Revolutions per minute, 906.

From these figures it is possible to deduce certain coefficients, one of which is the slip of the screw, which figures out at 11.5 percent. Another coefficient of performance, according to a French formula, and based on the assumption that the maximum of 45 horsepower was actually attained on trial, is as follows:

$$M = \sqrt[3]{\frac{B^2 V^3}{F}} = \sqrt[3]{\frac{0.27 \times (17.11)^3}{45}} = 3.11.$$

These results are remarkable, particularly in view of the fact



THE GENERAL ARRANGEMENT PLAN

that, according to a well-authenticated article on automobile boats appearing in 1904, in *le Bulletin de l'Association Technique Maritime,* this has never before been obtained by a boat of this type, not even by the celebrated French boats *Lutèce* and *Rapée III.*

These results are due in a large measure to the splendid form of the hull. At high speed, this hull raises itself forward and appears to glide over the water, producing at the stern a very slight wake. For several months the boat has been operating at full speed upon the narrow and crowded Meuse at Liège, without having provoked the slightest objections from the boatmen on account of any swell set up. The photograph of the boat at full speed, which we publish, will show this remarkable absence of wake.

The boat, covered with its hood, has shown itself to be a very good sea boat, even in rough water, while its stability is considerable. It has often carried as many as seven people, under which circumstances the speed has been reduced to 29 kilometers (15.6 knots) per hour. The rudder, in spite of its small dimensions, allows the boat to turn very sharply; at full speed the boat can easily turn around upon the Meuse at a place where the width is not greater than 100 meters (109 yards). In spite of the almost continuous and excessively hard service to which the boat has been subjected since last August, neither the hull, nor the engine foundations, nor the motor itself give the slightest trace of fatigue. This is the incontestible advantage accruing from a steel hull, particularly when fitted with a powerful engine.

Talks on Lubrication on Marine Engines.

Noon-hour talks over the dinner pail concerning the technicalities of engineering, mechanics, machine fittings, etc., are getting to be quite common. The other day the subject drifted along the lines of lubrication of bearings of marine engines, and the older and more experienced men gave us a few points on looking out for troubles, that were good. The spokesman made hand-chalked sketches on the shop floor, and some I believe are good enough to reproduce. The first sketch shows the necessity of keeping a cover on a bearing. It represents a shaft without a bearing, which is supplied with a means for being turned at about 100 revolutions per minute by the belt *C*. There is an oil-delivery tube at *A*, and drops of oil are deposited upon the turning shaft as shown. The movement of the surface of the shaft carries the films of oil along for a short way, but the centrifugal force is such that the oil flys off the surface at *B*. If, however, as in practice, there is a cover provided, this oil does not escape, but is confined close to the shaft.

Of course there are conditions that come in to make the parts heat and bind even when free lubrication is provided, as in Fig. 2. Here we show an unevenly worn journal (in exaggerated proportions), which wear is partly rectified by the insertion of specially-fitted sleeves or parts of sleeves, like *D*. The tapered condition of sleeves that are inserted like this often results in binding and increased friction, squeaking, and groaning. Sometimes one finds that the sleeve binding is out of line and that there are gaps, as at *E*. In the open space at right of the journal is given opportunity for oil and metal dust, flyings, waste, etc., to gather and gradually work into the lubricant and cause heating and binding in the bearing.

Then, again, I have run across bearings in which even the best grades of oil cannot keep the temperature down, and prevent squeaking. This was due entirely to the dropping of one end of the shaft in the worn journal, as represented in Fig. 3. This sketch shows the gaps at *F* and *G*, caused by the worn parts in the sleeves. New sleeves will overcome this trouble; but oftentimes the defect is allowed to run and the heating blamed to the grade of oil used.

The clogging of oil channels is often a source of trouble with journals. The channels get clogged by the oil drying and fine particles of dust collecting. I have seen several cases in which complaints were made of the undue heating of journals of very good engines, while upon examination the oil holes were found completely clogged. In one instance the parties purchasing a new engine for a yacht complained to the builders that the machine was running hot. An investigation proved that a poor grade of oil was used, and that the oil holes of the heated parts were nearly closed by the congealed oil, so that hardly a drop of the quantities of oil squirted over the caps, the machinery, and engine space reached the bearing surface; Fig. 4 shows the simple cause. The

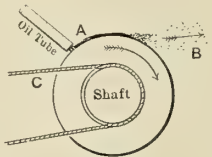


Fig.1.

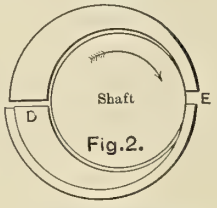


Fig.2.

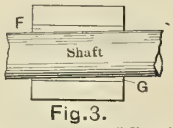


Fig.3.

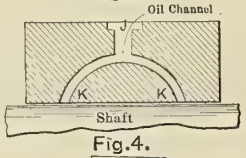


Fig.4.

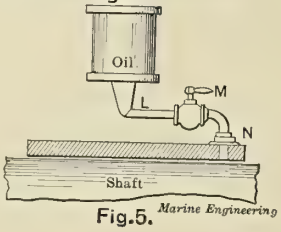


Fig.5.

oil feeds were through a channel with double ends, as at *K, K,* the feed being at *J*. The cheap lubricants refused to make these channels and collected in the neck, choking the passage, and thereafter the bulk of the oil with which the oiler drowned the parts simply flowed to the deck. We first soaked the journal and parts in lye water, to remove all gummed matter, then a proper adjustment of feed was made with the right grade of oil, and the bearings never heated again.

Fig. 5 is a drawing of a common form of oil feed employed in marine service, which often gives trouble. The reason is that the elbow *L* is too cramped and the oil passage readily chokes up, particularly when heavy oils are run. The valve at *M* ought to be examined to see if the passage is free. I have frequently found new valves leak at the very beginning, thus wasting the oil. In this form of feed the inlet to the bearing is sometimes at *N*, while the other end of the bearing is neglected. If these bearings are taken apart, nine times out of ten more oil will be found at the delivery side than the other. Often the discrimination is such that the side not supplied with oil is almost dry, and trouble results. It is a good plan to insert the feed tube in the center or have a tube for each end.

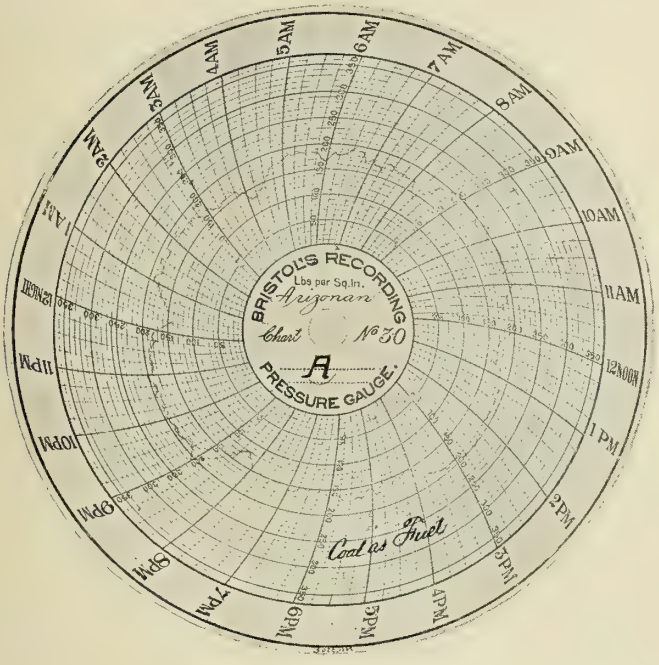
"Ex-Machinist."

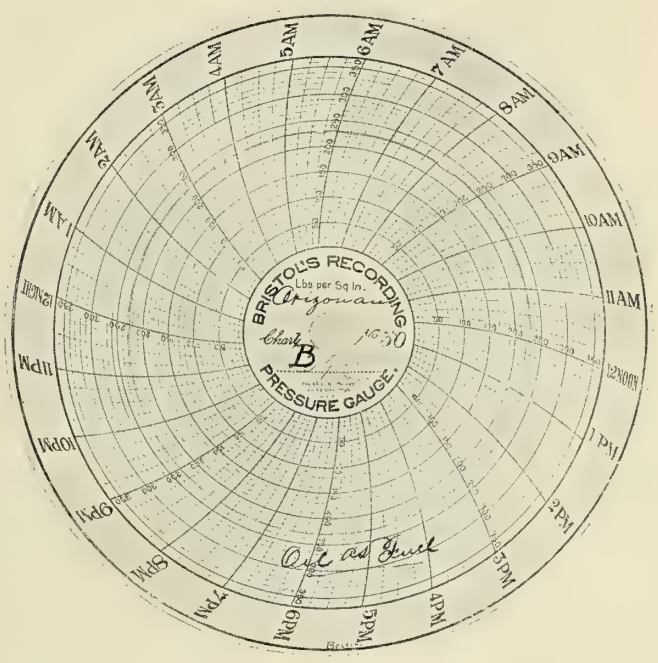
On page 125 of our March number was published a photograph of the new armored cruiser *Maryland,* taken by Mr. N. L. Stebbins, of Boston, during the trial trip of the ship, and copyrighted by him. Through an oversight due credit was not given Mr. Stebbins for this photograph, and we desire here to rectify the error.

Oil Fuel as Compared with Coal.

The many advantages of oil fuel on shipboard as compared with coal have frequently been mentioned in these columns, such as decreased weight, stowage in water bottom whereby bunker space is available for cargo, ease of loading, decrease in boiler-room force and expense of up-keep, etc. One important point—the facility of control whereby a high steam pressure may be maintained practically constant—is strikingly shown by a comparison of the two diagrams here presented, taken by a recording-pressure gauge on the steamer *Arizonan,* of the American Hawaiian Steamship Company.

Each chart is a record for twenty-four hours while the ship was at sea. The one marked *A* was taken on a voyage from San Francisco to New York when coal was burned in the boilers. So successful had been the results of the oil-burning apparatus on board the *Nebraskan,* another vessel of this fleet, that a few months ago the *Arizonan* was fitted with the Lassoe-Lovekin system, and on January 9, 1905, completed her first round trip from San Diego to New York and return with oil as fuel. It was on this voyage that chart *B* was made. On both charts the pencil of the gauge records 15 pounds above the actual.





The steam pressure when coal was burned did not exceed 185 pounds, while it dropped when cleaning fires as low as 105 pounds at 8:30 P.M. In fact, the drop at the beginning of each watch, when the fires were cleaned, is most noticeable, as well as the variations during watches. In contrast to this, with oil fuel, the pressure was maintained at 215 pounds throughout the day.

A word of explanation should here be given as to the cause of the great drop when burning coal. The *Arizonan* has scant boiler power, and with coal the boilers were at all times forced to their limit to supply steam, so that as soon as a fire was drawn, which means that that furnace was temporarily out of service, the steam went down at once. To have maintained, when burning coal, the same steam pressure as when oil was used, would have required fitting another boiler; thus with oil fuel the capacity of a boiler plant is increased about one-third, or a given power may be attained on 25 percent fewer boilers.

Another most important advantage resulting from the use of the oil-burning apparatus is the increased steam pressure; with coal the maximum was about 200 pounds, while with oil it is maintained uniformly at 215 pounds. The actual results on this ship meant an increase in speed of one knot—from 9 to 10 knots. The improved running of the main and auxiliary machinery due to constant steam pressure are recognized by any engineer; not only will less attention be needed in adjusting and running the main engines but the auxiliaries, such as the boiler feed pumps, etc., will naturally run at constant speed under uniform steam pressure.

The boilers themselves will require less repairs where oil is used, as the furnaces, back connections and tubes are subjected to practically the same temperature every hour of the day. As the oil combustion is complete no soot is deposited and the heating surfaces are clean and thus at their maximum efficiency for transmitting the heat. Where coal is used, each time the furnace door is opened a draft of cold air enters, striking the plates, and thus subjecting them to strains due to unequal temperature.

These practical and other economic advantages resulting from the installation of oil-burning equipment have been decidedly proven on four steamers of this company. The boiler dimensions of the *Arizonan,* a sister ship of the *Nebraskan,* and the data of her recent voyage from New York to San Diego are given as follows:

Number of boilers	3
Length inside heads	10 ft. 2 ins.
Diameter inside	14 ft. 3 ins.
Furnaces each	4
Combustion chambers, each	2
Furnaces, diameter inside	34 ins.
Grate surface as first fitted, bars, 4 feet 6 inches each	54 sq. ft.
Heating surface, each	2,040 sq. ft.
H. S. ÷ G. S.	37.7
Calorimeter area	9.8 sq. ft.
H. S. 3 boilers	6,120 sq. ft.
Draft average for voyage	26 ft. 8 3-4 ins.
Displacement for voyage	16,000 tons
Steaming time, 53 days, 11 hours, 52 minutes.	
Distance run	12,773 miles
Speed average	9.94 knots
Slip of screw	20.3 percent
I. H. P. main engines	2,700
I. H. P. total	2,900
Oil used, Texas reduced	13,418 bbls.
Average consumption oil per hour	10.45 bbl.
Average consumption oil per I. H. P. hour	1.17 lbs.

The oil consumed was measured by meter at a temperature of 100 degrees Fahrenheit. "F. D. H."

Marine Engineering

Published Monthly by

MARINE ENGINEERING

INCORPORATED.

17 Battery Place, - - NEW YORK

12 St. Benet Place, Gracechurch St., LONDON, E. C.

H. L. ALDRICH, President and Treasurer

PROF. W. F. DURAND, Advisory Editor

SIDNEY GRAVES KOON, Editor

GEORGE SLATE,
Vice-President and Advertising Representative

Branch Offices. Philadelphia, Pa., Machinery Dept., The Bourse, S. W. Anness.
Boston, Mass., 170 Summer St., S. I. Carpenter.

TERMS OF SUBSCRIPTION.

	Per Year.	Per Copy.
United States, Canada and Mexico	$2.00	20 cents
Other countries in Postal Union	10/6	1/-

Entered at New York Post Office as second-class matter.

Notice to Advertisers.

Changes to be made in copy, or in orders for advertisements, must be in our hands not later than the 15th of the month, to insure the carrying out of such instructions in the issue of the month following.

The New York Motor Boat Show.

With the great resources of Madison Square Garden taxed to the limit by the hundreds of exhibits, the National Motor Boat and Sportmen's Show of 1905 was opened on the evening of February 21st and closed just before midnight on the 8th of March. By a gradual process of evolution, a sportsmen's show, with motor boat annex, has now become in effect a motor boat show, with the sportsmen as auxiliaries. The rapid progress and development in the design of internal combustion motors, together with the concurrent introduction of innovations in the forms of hulls, renders such an exhibition of great value to the student of the subject and to the engineer; while the opportunity afforded intending purchasers to see under one roof such a great variety of types and makes, of sizes and styles, not only aids them in making a satisfactory choice, but arouses in many cases such an interest in the subject among those who attend the show as mere onlookers as to procure other purchasers from this class, and thus extend the general scope of the business among the purchasing public. As a financial venture, therefore, the annual exhibition is regarded as a success, and exhibitors feel well repaid for the great expenses and labor incident to its inception and realization.

The possibilities afforded by the artificial lagoon to demonstrate the qualities of certain of the boats and motors under exhibit are of very necessity curtailed by the rigid limitations in the size of this body of water. It does constitute, nevertheless, a most useful adjunct to the show, and the numerous boats floating upon its surface, and occasionally making a circuit of the central island, offer valuable information to the observer of the operation and control of both engines and boats. The necessarily frequent stopping and turning, with an occasional backing movement, demonstrate clearly the maneuvering qualities of the boats under operation, and form a splendid opportunity to observe their behavior under a variety of conditions.

Altogether, the show may be said to have been a huge success. Thousands of visitors were accommodated; thousands of dollars' worth of business was transacted, not only between the exhibitors and the public, but among and between the manufacturers themselves; while the prospective business, not actually concluded during the show, has amounted to thousands of dollars more.

A New Era in Steam Navigation.

With the recent arrival of the Cunarder *Caronia,* and the completion of the maiden voyage of the Allan Line steamer *Victorian,* a new era in the history of transatlantic steam navigation may be said to have opened. The *Caronia* marks the transition from the old to the new, not from being essentially different from preceding ships, which she is not, but because of the fact that she is one of a pair of ships, and that her sister, the *Carmania,* whose first trip it is expected will be made early next fall, will be one of the new type, in so far as her method of propulsion is concerned, that is to say, in the main point of divergence between the two types. The hulls of these two ships are as nearly as may be identical in form, though that of the *Carmania* is considerably "fined" aft to admit of the substitution of the three turbine-propelled screws in place of the usual two propellers actuated by reciprocating engines, as fitted in her sister. The designed power being the same, these two ships will afford a most excellent basis for an exhaustive and instructive comparison between the two forms of propulsion, upon so large a scale as to be of the utmost value to the Cunard Company itself, and to the engineering world in general. Followed, as they are shortly to be, by the two mammoth flyers which this company has under construction, it is seen that a very large wedge has been inserted into the future field of steam propulsion.

The Allan Line steamer *Victorian* is to be followed at an early date by her sister *Virginian,* of similar size and power, which is already in the water. Both ships are fitted with steam turbines, and are expected to reduce the time from Liverpool to Montreal by nearly an entire day, as compared with the present running schedule. While not affording the same sort of a basis for comparison as with the two new Cunarders, there will be presented a splendid opportunity for observation, and for the compilation of an immense amount of such data as will be useful for constructive purposes in subsequent designs.

While recognizing the utter futility of discounting the future, particularly in the matter of engineering progress, and in these exceedingly inventive and strenuous days, it may yet be said to be highly improbable that any innovation in marine propulsion will ever rival, in far-reaching effects, the introduction of steam as a propelling agent. Comparable, however, with the general substitution of screw propellers for the older paddle wheels, with the successive economies resulting from the displacement as a constructive material of wood by iron and of iron by steel, the possibilites opened up by the advent of the steam turbine are little short of marvelous, when viewed from the standpoint of the marine engineering practice of a decade since. On the side of the engineer, the turbine effects a remarkable saving in the weight required and in the space occupied for a given power; for continuous running at or near its designed capacity it offers a very considerable economy of operative expenses chargeable to fuel, while under all conditions a large economy ensues in the matter of attendance; the high rate of revolution of the turbine and its propeller makes possible the utilization of a small diameter in the latter, thus effecting at one and the same time an economy in weight and a sufficient submergence beneath the surface of the sea to furnish reasonable security against any such thing as "racing," while the mass of the turbine itself, acting virtually as a fly wheel, would operate as an additional damper upon any tendency of the apparatus to run away. From the viewpoint of the naval architect we have the possibilities of the installation of increased cabin accommodations on account of the decrease in space occupied by machinery. The military naval engineer is interested in the fact that the essential construction of the turbine is such as to place the entire engine close around the shaft, increasing stability and greatly decreasing the head-room required, and thus placing the prime mover far enough below the water-line to be amply safe against the fire of an enemy at sea. Lastly, the passenger on the ship fitted with the turbine method of propulsion is given the novelty of a passage nearly or entirely free from the vibration always a feature of heavy reciprocating engines, however well balanced they may be.

Looking a little into the future, it appears more than probable that the gasoline engine, which has been making enormous strides in adaptability and reliability during the past five or six years, due largely at first to its ultilization as an agent of operation for automobiles, and later quite as much to its rapid introduction as a propelling device for pleasure craft of all descriptions, but more particularly those of small size, will become more and more available for larger powers than those for which it is now usually fitted; will perhaps gradually displace the steam engine for vessels making commercial trips of no great length, such as fishing and coasting craft, river boats carrying both passengers and freight, and possibly some of the smaller types of ocean steamers. Nothing at present in sight would appear to justify the assertion, however, that it will soon become available for the larger class of vessels.

Another form of installation now attracting considerable attention from engineeers is that of producer plants in conjunction with gas engines. This combination is said to have a considerable future before it, in the larger sizes of vessels; while the gasoline engine, *per se*, using the liquid form of the fuel rather than the gaseous, takes care of those of smaller dimension. This is as yet, however, entirely in the experimental stage, and cannot be considered as an immediate factor in the question.

All of these changes, if changes they are to be, are of great importance, but, as in the case of the accomplished introduction of the steam turbine, cannot be said to be revolutional in character, as was that of the steam engine, nearly a century ago. The motors proposed are all heat engines; all operate one or more screw propellers, as has been largely the case for the past fifty years; all types and forms are reciprocating in character, the steam turbine alone excepted (for the much-heralded gas turbine has thus far attained no definite status as an agent of propulsion); all depend for their efficient performance upon the coincident operation of a certain number of auxiliary devices and appliances, much the same in character for the varying types; and, lastly, all call for the consumption of large quantities of fuel, which must be carried on the voyage, and which cut down, to the extent of their own weight, the amount of cargo which can be transported.

Putting aside, therefore, all idea of changes of an extremely radical character, it will be seen that the field of future improvements becomes narrowed to the question of obtaining here and there small economies in operation—here a waste (heat or otherwise, but always, in the limit, heat) decreased or cut off entirely; there a device introduced to improve the effectiveness of some part of the total entity which may, perhaps, have been previously overlooked as of too small importance in the whole scheme to merit serious consideration. And from this gradual refining process may be expected most of the improvements of the immediate future.

The Screw Propeller and Marine Machinery.

With this issue we conclude the exhaustive discussion upon the design and operation of the screw propeller, from the pen of Professor Durand, which has run through four numbers. The importance of possessing accurate information upon such a subject is very great indeed, and is rapidly becoming more so with the introduction of higher rates of speed and of revolution. One of the main avenues for refinement in economy of operation lies right here, and the question is destined to receive a large measure of attention in the near future.

We hope to resume, in June, the series of articles by Professor Durand upon the Design of Marine Machinery, which was interrupted more than a year ago by pressure of work falling upon the author.

ABSTRACTS FROM RULES AND REGULATIONS OF THE AMERICAN BUREAU OF SHIPPING.

EDITION OF 1904.

ARRANGED BY HARRISON S. TAFT.

PART II.

17. Deck Plating, Deck Stringers and Tie Plates.

1. The deck stringers and deck plating should maintain their midship scantling for half length amidships.

2. Steamers not requiring a complete or partial steel deck must have a steel deck of at least 10 pounds worked over the boiler and engine space. Said deck is to extend one beam space fore and aft of said space and thence taper into the deck stringer.

3. Coal bunkers located between the boiler and engine rooms are to have a complete steel deck worked over the same.

4. Partial steel decks should taper for at least 6 frame spaces beyond their required length of 1-2 length amidships into the stringer and tie plates.

5. Steel decks should be worked continuously through all bulkheads.

6. The plating at the mast partners, large deck openings, and at all heavy fittings should equal the weight of the deck stringers amidships.

TABLE 7.—TIE PLATES AT LARGE DECK OPENINGS.

Length of Opening.	Width of Tie Plates.	Notes.
12 feet under 16 feet	1½ times regular width	Tie plates of extra width are to taper for 3 frame spaces into regular width.
16 " " 20 "	Twice regular width	
20 " and above	a. Twice regular width and the deck plated over in wake of said opening with 10 lbs. plate.	a. Said plating is to extend one beam space beyond each end of said opening.

7. When no steel deck is laid tie plates are to be fitted on each side of the middle line on each deck, so as to take the coamings of hatches, etc. Diagonal tie plates are to be worked on the upper decks of all sailing vessels not having a steel deck, radiating from mast partners to stringers, to which they are to be double riveted. Discontinuous tie plates are to lap each other at least one beam space.

8. Mast Partners. They should extend fore and aft 2 beam spaces for alternate spaced beams, and 4 beam spaces for every frame-spaced beam. They should be three times the diameter of the mast hole in width and of the same weight as the deck stringer amidships. The coaming rings are to be of the same scantling as the midship frames.

18. Hold Stanchions.

1. Said stanchions should be fitted at the center line to every beam for 3-5 length amidship, and on alternate beams fore and aft of same. They should have at least two rivets in each end. The hold and the deck stanchions should be placed over each other.

2. When beams are fitted to every frame the stanchions are to be fitted at alternate beams to a strong back fitted under said beams and clipped to same.

TABLE 8.—SPACING OF QUARTER STANCHIONS ON EACH SIDE OF CENTER-LINE STANCHIONS.

Vessel's beam	44 feet under 56 feet	56 feet and over	Sailing vessels over 40 feet.
Spacing of stanchions	On alternate beams	On every beam	On alternate beams for $\frac{3}{5}$ L.

When vessel's beam is under 44 feet only the center row of stanchions need be fitted.

3. Stanchions are to be fitted to all beams under the windlass, capstan, deck engine, etc., and under the fore-and-afters of hatchways over 6 frame spaces in length.

4. Masts stepped on a deck are to have extra stanchions fitted to the three deck beams under said step.

5. Hold stanchions over 14 feet in length, and deck stanchions over 8 feet in length, are to be made respectively 1-2 and 1-4 of an inch larger in diameter than otherwise.

6. Bulwark Stanchions. They should be spaced not more than 6 feet apart and are to be from 1 1-4 inches to 2 inches in diameter. They should have spurs to the bulwarks, with doubling plate at same.

19. Bulkheads.

1. There shall be a transverse water-tight bulkhead at a suitable distance forward of the stern (except in paddlewheel steamers), with a water-tight flat extending aft to the stern so as to form the aft peak.

2. There shall be a transverse water-tight bulkhead at each end of the machinery space.

3. All of the above bulkheads should extend to the upper deck in all vessels; but to the second deck in hurricane-deck vessels. In three-deck vessels the aft peak bulkhead is to extend to the second deck only.

4. When the cargo space forward or aft of the machinery space exceeds 75 feet in length said spaces shall be divided by an intermediate water-tight bulkhead at the middle of each space. If said cargo spaces exceed 140 feet two intermediate bulkheads shall be worked in each space. Said bulkheads are to extend to the deck next above the deck load water-line.

5. All water-tight bulkheads shall have double frame angles with double top angles connection to the deck plating, and to the inner bottom plating when worked.

6. Stiffeners equal to the midship frames are to be worked vertically on one side not more than 24 inches apart; horizontally on the other not more than 48 inches apart. The horizontal stiffeners shall be bracketed to the shell.

The vertical stiffeners shall be well secured to the floor plates, but bracketed to the inner bottom plating when used. The brackets are to equal the thickness of the flange of the angle bar to which they are riveted.

7. All horizontal stiffeners on the collision bulkheads, and on the lower parts of all other bulkheads, 40 feet or more in width, shall be of bulb angle of specified weight. Vessels of other than American registry can have the vertical stiffeners on the water-tight bulkheads spaced 30 inches apart.

8. *a.* Vessels with hold beams, deep frames, or web frames in lieu of hold beams, of less than 40 feet in breadth, should have the alternate stiffeners on all bulkheads either of deep frames with reverse bar, of bulb angle, of channels, or of Z bars, of specified scantling.

b. When the breadth is 40 feet and under 48 feet, or depth from top of floors to top of the lowest tier of regular deck beams amidships is over 20 feet, all bulkheads should have all their horizontal and vertical stiffeners of the foregoing scantling.

c. When the breadth is 48 feet or over, in addition to the above, all bulkheads are to have, midway top of floors and the lowest tier of deck beams, a girder beam of plate, one frame space wide, with an auxiliary beam supporting its outer edge.

Said plate girders are to be clipped to the bulkhead plating between the vertical stiffeners, and have a continuous face bar clipped to the face of said stiffeners. Said plate girders are to be treble riveted to the side stringers, and have bracket plates, equal to the thickness of the girder plate, single clipped to said girder at alternate stiffeners.

9. Sailing vessels need have but the two end collision bulkheads.

TABLE 9.—LOCATION OF FORWARD COLLISION WATER-TIGHT BULKHEAD.

Length of ship in feet.....	Under 200	200 under 350	350 under 500	500 and above.
Distance aft of stem in portion of vessel's length.....	$\frac{1}{10}$ L	$\frac{1}{12}$ L	$\frac{1}{15}$ L	$\frac{1}{16}$ L

10. Center line bulkheads shall have stiffeners equal to the midship frames spaced at every frame. Alternate stiffeners shall be secured to the deck beams and be bracketed at the bottom. The plating is to have double top and bottom angle bar connection of the same scantling as the reverse bars, but they need not exceed 3 1-2 by 3 1-2 inches.

11. Coal Bunker Bulkheads. The plating is to be 2 pounds less in weight than used for the shaft tunnels. The fore-and-aft bulkheads are to have stiffeners spaced 30 inches apart, of the same flange size, but 2 pounds lighter than reverse bars. Every third stiffener is to be stayed to the side framing. The plating is to have a single angle bar of stiffener size at the top and bottom. Athwart ship bulkheads are to have stiffeners of the same scantling as the reverse bars, spaced 30 inches apart, having single top and bottom bars of the same size as the stiffeners, but not over 4 inches by 3 inches, when said bulkhead is not located on a frame.

12. Bulkheads at the front of a full poop and enclosed bridge houses are to be of 10-pound plate when the second number is under 16,500, but of 12-pound plate when the second number is or over 16,500. The height of the coaming plate above the planking is to equal the depth of the main deck beams, and it is to be 2 pounds heavier than the above plating. When the breadth of a vessel is 30 feet the stiffeners are to equal the frames and be spaced 30 inches apart. When said breadth is 30 and under 40 feet, alternate stiffeners are to be doubled. When said breadth is 40 feet of over all stiffeners are to be doubled. All stiffeners are to be bracketed to the deck plating. Said bulkheads should be located over a main deck beam.

13. Doors will not be allowed through a water-tight bulkhead in the lower holds, or through the front of an enclosed bridge house that abuts to a raised quarter deck; nor companion ways in the bulkheads in the front of a raised quarter deck.

20. Boiler Saddles.

Boiler saddles, at the point of their contact with the hull, should equal the diameter of the boiler.

21. Shaft Tunnels.

Second Number.	Weight of Plating.	Notes.
Under 10,000....	Not less than 10 lbs...	Top of tunnel under hatches is to be worked double or be covered with 2 inches of wood.
10,000 " 16,500....	" " " 12 "	Stiffeners equal to reverse bars are to be spaced not over 4 feet apart.
16,500 and above.....	" " " 14 "	To have a W T door at engine room end, which shall work from upper deck.

22. Breast Hooks.

They must be fitted between decks of 7 feet or more in height. In the lower holds or peaks they must not be more than 4 feet apart. They should be clipped to the shell, and have a beam at every frame. They are to be 2 pounds thicker than the side stringers at their ends.

23. Shell Plating.

1. The plates should not exceed 66 inches in width. They should maintain their midship weight for 1-2 length amidships. The plates should not be less than 6 frame spaces in length, except at the hoods, where if shorter than 5 frame spaces they are to be connected to the plates before and abaft same with treble riveted butts.

2. The butts in adjoining strakes are to have at least a two-frame space shift; those in alternate strakes at least a one-frame space shift. The garboard butts must be kept clear of the keel scarphs and have a clear shift of

two frame spaces from each other on the opposite sides. All sheer strake butts are to be shifted at least two frame spaces from the butts of their adjoining deck stringers.

3. When a double bottom is fitted the shell plating within range of said bottom, including the flat keel plates and garboards, may be 2 pounds less in weight than otherwise called for.

4. When the boilers and engines are placed aft the upper deck stringer, sheer strake and top sides shall maintain their midship weight throughout the machinery space, with treble riveted butts.

5. The boss plates and garboard strake plates secured to the propeller post in single-screw vessels are to be of the same weight as their respective strakes amidships.

When the second number is 19,000 or above, all plates connected to the propeller post are to be of the same weight as their respective strakes amidships.

6. Sailing vessels whose second number is 16,500 and under 25,000 should have two bilge strakes, all fore and aft, 2 pounds heavier than otherwise. When said number is 25,000 or above, three bilge strakes, all fore and aft, should be 3 pounds heavier than otherwise.

TABLE 10.—BULWARK PLATING.

Second Number.	Under 8,000	8,000 not over 16,500	Over 16,500
Weight of..... Bulwark plating....	Not less than 6 lbs..... 8 lbs. if over 3 feet high....	8 lbs.	10 lbs. 8 lbs. if less than 3 feet high.

Length of bulwark plates should be a multiple of stanchion spacing so that said stanchions may come at a butt of said bulwark plates.

7. If the holes in the sheer strake for side port lights which are located within the range of half the vessel's length amidships are or exceed 9 inches in diameter, the sheer strake is to be reinforced at said lights.

8. When single butt straps of a main sheer strake are worked on the inside and are cut at the upper deck stringer, the top of the sheer strake must extend above the gunwale bar so as to allow of two horizontal rows of rivets for said strap worked above the deck stringer gunwale bar, said strap being joggled and fitted into the throat of the gunwale bar and riveted thereto.

9. The main sheer strake must not be cut for a freight or coal port.

TABLE 11.—MINIMUM WIDTH IN INCHES OF KEEL, GARBOARD, AND SHEER STRAKES.

Second Number.	Under 10,000	10,000 under 16,500	16,500 under 25,000	25,000 under 48,000	48,000 under 80,000	80,000 and above.
Flat keel plate.....	27	30	33	36	56	36
Garboard strake.....	30	33	36	39	42	45
Sheer strake.....	33	36	40	44	48	54

For ⅗ length amidships.

24. Freight Ports.

When cut in the sides of the ship between decks (below upper deck sheer strakes), if of more than two frame spaces in width, said ports must have web frames at each end, extending from deck to deck, and connected to same with double clips. The shell strakes above and below said openings must be doubled for three frame spaces fore and aft of said doors.

25. Butt Straps.

1. Single butt straps must be at least 2 pounds heavier than the plates they secure; but when triple riveted 4 pounds heavier than the plates they secure. The thicker of the two plates is to determine the weight of said strap.

2. Double butt straps must not be less than 3-4 the thickness of the plates they secure.

3. Butt straps of the flat keel plates are to be at least 6 pounds heavier than plates they secure.

4. Butt straps of floors, web frames, all continuous keels and keelsons must be worked double.

5. Bosom pieces to angle bar butts must be of the same thickness as the bars they secure.

6. Butts of the angle bars of the floor or bilge double angle bar keelsons are to have a 12-frame space shift of each other.

7. Butts of the plate side stringers or their angles must not come at a web frame.

8. Butt straps of all shell plates 54 inches or more in width are to be at least 5 pounds heavier than the plates they secure.

9. Butt straps of an upper deck stringer (second deck in hurricane-deck vessels) 54 inches or more in width should be worked double; but when the stringer is worked double, said butt straps can be worked single.

10. Shell Plating and Deck Stringers. See Table 12.

11. Single butt straps of the upper deck sheer strake, second deck in hurricane-deck vessels, when equal in width to and fitted on the inside of said strake, are to have the gunwale bar joggled around them, or else liners fitted between said bar and the sheer strake plate. When said butt straps are cut at the deck stringer, the part of said straps above the deck stringer are to have two horizontal rows of rivets through the sheer strake above the gunwale bar, and they shall joggle over into throat of said bar and be riveted thereto.

12. Shell doubling plates are to have butt straps 4 pounds heavier than said doubling plate.

13. Butt straps of all masts and spars are to be 3 pounds heavier than the plates they secure.

14. Bulb plates at their butts should have the bulb cut off on each side so as to allow a full width butt strap.

15. Single butt straps of deck ties, stringers, and deck plating should be fitted on under side.

26. Riveting.

1. With plates of different thickness the thicker of the two is to regulate the diameter of the rivets.

2. Tap rivets should be 1-8 of an inch larger in diameter than the corresponding ordinary rivet.

3. Bar keels, stems, and stern posts should be double zigzag riveted. The rivets should be 1-4 of an inch larger in diameter than those used in riveting the garboards to their adjoining strakes.

4. With a vertical center line keel and keelson the garboards must be double riveted to same, with an intermediate row securing the side plates to the center line keelson plate before the garboard plates are put on.

5. Flat plate keels; floor plates; web frames; horizontal floor stringer plates; all continuous keels, keelsons, and rider plates must have treble riveted butts throughout.

6. The laps or edges of water-tight bulkheads and the longitudinal laps or seams of steel decks should be single riveted. Vessels whose second number is or above 14,000 are to have their bulkhead plating double riveted to the floors.

7. The longitudinal laps of all shell plating under 16 pounds worked in and out (those of lower edge of sheer strake under 14 pounds) are to be single riveted; but when of 16 and 14 pounds respectively and above, said seams are to be double riveted. The type of midship riveting is to be carried all fore and aft.

8. The flush longitudinal seams of all shell plating under 20 pounds are to have single riveted edge strips; but for 20-pound plating and above, double riveted edge strips.

9. All double riveting of longitudinal laps and seams is to be chain fashion.

10. All butts of shell plating, deck stringers, and deck plating should be at least double riveted.

11. The lapped butts of shell plating for vessels whose second number is under 16,500, are to be treble riveted for 1-2 length amidships and double riveted at ends. If said number is or over 16,500, said butts are to be treble riveted throughout.

12. Butt straps of all shell plates 54 inches or more in width are to be treble riveted.

13. Shell Plating and Deck Stringers.

TABLE 12.—BUTT STRAPS AND RIVETING SCHEDULE FOR SHELL PLATING AND DECK STRINGERS.

Butt straps of:	Second Number.	10,000 under 16,500	16,500 under 22,000	22,000 under 30,000	30,000 under 48,000	48,000 and above
	Sheer strake	T R+4 Lbs.—½ L	T R+5 Lbs.—½ L	T R+6 Lbs.—½ L	T R+7 Lbs.—½ L	T R T+10 Lbs.—½ L
	First bilge strake	T R+4 Lbs.—½ L	T R+5 Lbs.—½ L	T R+6 Lbs.—½ L	T R+7 Lbs.—½ L	T R T+10 Lbs.—½ L
	Second " "	D R+2 Lbs.	T R+5 Lbs.—½ L	T R+6 Lbs.—½ L	T R+7 Lbs.—½ L	T R T+10 Lbs.—½ L
	Remaining "	D R+2 Lbs. T H T	D R+2 Lbs. T H T	T R+6 Lbs.—½ L	T R+7 Lbs.—½ L	T R T+10 Lbs.—½ L 5 Lbs. at ends
	a. Upper deck stringer	D R+2 Lbs. T H T	D R+2 Lbs. T H T	D R+2 Lbs. T H T	T R+7 Lbs.—½ L	T R T+7 Lbs. T H T
	Lower " "	D R+2 Lbs. T H T	D R+2 Lbs. T H T	D R+2 Lbs. T H T	D R+2 Lbs. T H T	T R+7 Lbs.—½ L
	Shell plates over 54" wide	T R+5 Lbs.	T R+5 Lbs.	T R+5 Lbs.	T R+10 Lbs.	1 Lb. heavier than otherwise
	Deck stringers over 54" wide	D R+double straps	D R+double straps	D R+double straps	T R+10 Lbs.	T R+double straps

a. Second deck in hurricane-deck vessels.
D R = double riveted; T R = triple riveted; T H T = throughout; T R T = triple riveted throughout.
Plus sign (+) means "with." Minus (—) means "for."
Example; T R + 4 lbs. — ½ L means: triple riveted butts with straps 4 lbs. heavier than the plates they secure for ½ length amidship.
All the above triple riveted butts to have three complete rows of rivets.
Flat keel plates are to be triple riveted throughout and have straps 6 lbs. heavier than the keel plates.

14. In three-deck vessels, the butt straps of the sheer strakes, of the upper-deck stringer, and of at least two bilge strakes should be triple riveted for 3-5 length amidships.

15. When the engines and boilers are placed aft, the butts of the upper deck stringer, of the sheer strake and top sides should be at least treble riveted throughout the machinery space.

16. The two shell butts fore and aft of a shell doubling plate should be treble riveted. The doubling plates are to have treble riveted butt straps.

17. The longitudinal seams or edges of the inner bottom plating should be single riveted, except the laps or edges of the middle line strake, which are to be double riveted to the adjoining strakes. All athwartship butts are to be double riveted throughout.

18. Hatches on the upper decks and on the second deck of hurricane-deck vessels, and deck houses fitted over a deck opening, should have their coaming plate butts treble riveted.

19. Deck tie plates, laps, and butts should be double riveted.

20. (*a*) The seams or laps of all lower masts should be double riveted. If the length of the mast from the partners to the hounds exceeds 45 feet all butts above the partners are to be treble riveted. If said length does not exceed 45 feet only the butts of the lower half above the partners need be treble riveted, those in said upper half being double riveted.

(*b*) When the lower mast and top mast is made in one length the seams or laps may be single riveted; with double riveted butts throughout.

(*c*) The seams or laps of the topmasts and of the yards should be single riveted; with treble riveted butts throughout.

(*d*) The seams or laps of the bowsprits must be double riveted; butts outside of the gammoning are to be treble riveted, with those inside double riveted.

21. Angle bosom pieces should have at least 3 rivets on each side of their butts.

22. Keelson floor clips should have at least a 3-rivet connection to all floor plates.

23. Keelson rivets passing through four thicknesses of plate are to be 1-8 of an inch larger than otherwise.

24. Plug-neck rivets should be used in the shell plates, deck stringers, tie plates, and inner bottom plating.

25. The maximum weight of the shell plating outside of the sheer and garboard strakes is to determine the

size of the shell frame rivets. The frame rivets for the garboards and the sheer strakes are to be of the same diameter as for other strakes. The size of the frame rivets in the side plating above the second deck sheer strake in hurricane-deck vessels is to regulated by the thickness of the frame's web.

26. Surfaces of all plates and angles should be well painted before being riveted together.

TABLE 13.—COUNTER SINK ANGLES FOR RIVETS.

Diameter of rivet in inches.....	¼ to ⅝ inc.	¾ to ⅞ inc.	1 to 1¼ inc.	Above 1¼
Angle of countersink.....	53°	45°	37°	32°

The countersink is to extend to within $\frac{1}{32}$ of the depth of the plate or the flange of the angle bar.

Countersinks for tap rivets are to have a maximum diameter of from 1.4 to 1.5 times the diameter of the rivet shank: Depth of same is to be from .4 to .5 times diameter of said shank.

27. Double Bottoms.

1. When the frames are cut at the margin plates, frame brackets are to be fitted on both sides of the margin plate at every frame, when the floor plates do not extend to the inner bottom plating. But when the floor plates do extend to the inner bottom plating, aforesaid frame brackets are to be fitted at outside of the margin plate to those frames only on which the floor plates are worked

TABLE 14.—FRAME AND FLOOR BRACKET CLIPS. SPACING OF GUSSET PLATES.

Second Number.	Under 30,000	30,000 and Above.	26,000 Under 42,000.	42,000 and Above.
Said clips to margin plate....	a. Single.	Double.	.....	.....
Spacing of gusset plates.....	.....	.....	Every third frame.	Alternate frames.

a. Single clips are to be worked opposite to each other.

Gusset plates are to equal thickness of the inner bottom plating, and be worked throughout the inner bottom space.

A continuous plate stringer may be worked in lieu of the gusset plates.

2. The plating should be worked fore and aft. When worked flush the edge strips should be fitted on the upper side; butt straps on the under side. When a wooden ceiling is laid on an inner bottom the butt straps can be located on the upper side.

3. Longitudinal Girders on Top of Floors.

Said girders should be spaced not more than 40 inches apart. They are to be at least 15 inches high, with single top and bottom angle bars. Swash plates of 10 to 12 pounds are to be worked in way of the longitudinal midway between the center line and bilge. When the middle line girder on top of floors is worked in connection with an intercostal or continuous center line plate keelson, it may be made lighter than otherwise.

4. Cellular Bottoms with Continuous Longitudinals.

(*a*) Such bottoms should have a continuous center line plate keelson with continuous side longitudinals spaced not more than 5 feet apart. When the second number is under 10,000, floor brackets are to be fitted on both sides of the longitudinal girders at the top and bottom on alternate frames fore and aft of the machinery space. When said number is 10,000 and under 50,000, solid manhole floor plates are to be fitted at every fourth frame with the above floor bracket plates at the frames midway between said solid floors. When said number is 50,000 or above, the solid manhole floor plates are to be fitted on alternate frames.

(*b*) Under the engine bed solid manhole floor plates are to be fitted on every frame, but on alternate frames under the boilers and other machinery space.

(*c*) The top longitudinal bars are to be single and worked continuously. The bottom bars are to be single, intercostal between the frames. The solid floors and regular brackets are to be double clipped to the center vertical keelson; the intermediate brackets are to have single clips to same. Double bars are to be used throughout in way of the engine and boiler foundations.

5. Cellular Bottoms with Continuous Floors.

(*a*) Fore and aft of the machinery space the floors, double clipped to the center vertical keelson, can be worked on alternate frames if longitudinal bracket plates are worked at top and bottom on each side of said floors. If the second number is not over 18,000, and solid manhole intercostal longitudinal plates are used, the solid floors can be placed on every third frame. If said number is over 18,000, solid floors, fitted with solid manhole intercostal longitudinal plates, should be worked on alternate frames. The longitudinal brackets or the solid manhole floor plates are to have a single clip connection to the floors, shell, and inner bottom plating. Said brackets and floor plates should be spaced not more than 5 feet apart athwartships.

(*b*) Under the engine bed and boilers, the same construction as stated under (*27-4-b*) is to be used.

(*c*) Manholes should be cut in all solid non-water-tight floors between each side longitudinal.

6. When double bottoms are built with solid manhole floors on every frame, said floors shall extend in one piece from the center line to the margin plate, and from the shell to the inner bottom plating, with an intercostal longitudinal plate girder about midway from the center line to the margin plate. Said intercostal girder is to be clipped to floors and shell with angle bars equal to the reverse frames, and have a continuous angle bar on top. Said floors are to be double clipped to the center vertical keelson.

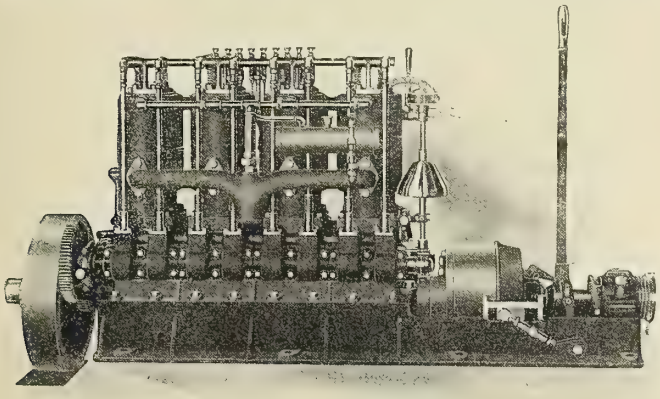
(To be continued.)

ENGINEERING SPECIALTIES.

New Lacy Gasoline Engine.

We illustrate herewith a new type of the Lacy gasoline engine, manufactured by The Brown-Cochran Company, Lorain, Ohio. Many features of decided merit are incorporated in this engine, among which may be mentioned accessibility, perfect control, quiet running, and freedom from adjustments. Quality and efficiency have been considerations of first importance. Valves are placed directly in the head of the cylinder and are set in cages which may be readily removed by simply loosening two nuts. Special attention has been given the mechanism for operating the valves. Cams are large and so designed as to do their work with but little noise.

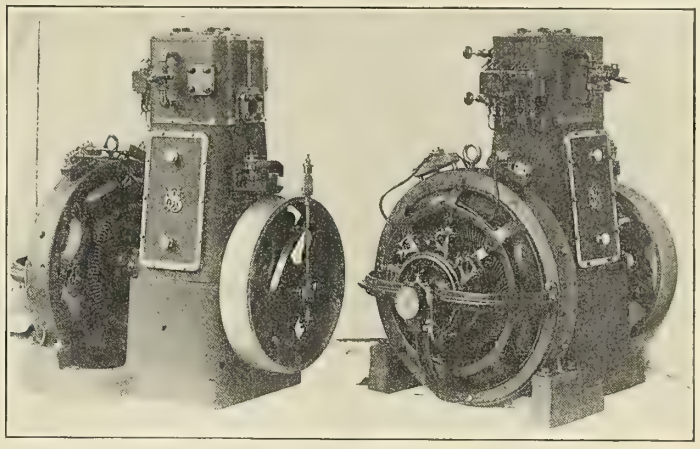
The carburetor is of the float-feed type and furnishes a uniform mixture at all speeds. An efficient governor controls the throttle valve and a hand lever is provided for controlling when desirable independent of the governor. Ignition is by jump-spark, or make-and-break, at the option of the purchaser. In either case the most approved devices are employed. Great care has been taken to produce a serviceable and efficient reversing gear. Ample proportions have been used and materials selected which would stand hard usage. The clutch is of the multiple plate type. Friction surfaces are large, avoiding frequent adjustment. It is impossible for the clutch to seize. The thrust bearing is so designed that no end motion can be worn into the parts of the reversing gear or crank shaft.



A point worthy of consideration is that all controlling levers are arranged to be conveniently reached by the operator standing at the reversing lever. The matter of oiling has been carefully considered and a very satisfactory system developed. A multiple feed oiler furnishes lubrication to all bearings subjected to hard wear and to the cylinders. This oiler is provided with eight feed glasses of large size. Bearings throughout are liberal, thus insuring long life, even with most severe service.

A New Sturtevant Generating Set.

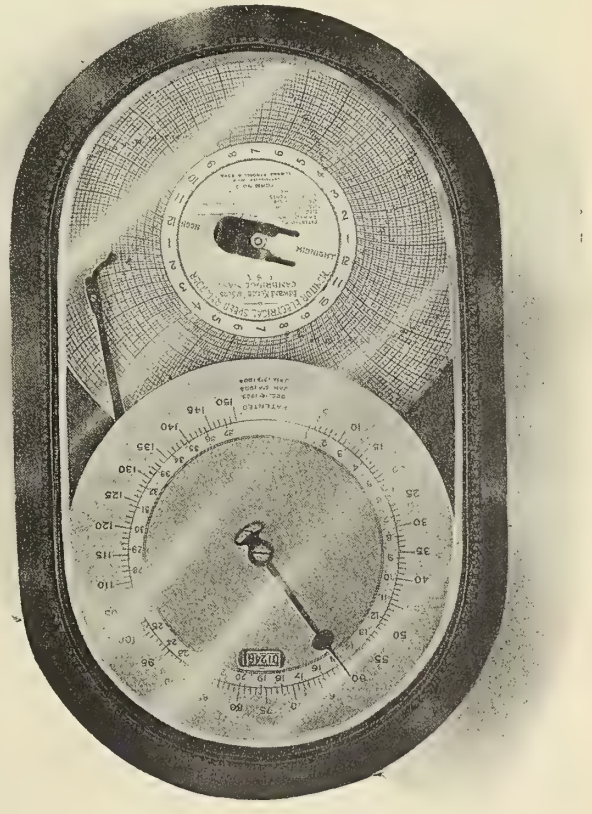
Originally built by the B. F. Sturtevant Company, of Boston, Mass., for the use of the United States government, and in accordance with specifications of the Bureau of Equipment, this generating set possesses features of special interest. The engine is of the high-speed reciprocating type, designed to run at 650 revolutions per minute. The cylinder, which is cast with the frame, is 4 1-2 inches diameter and 4 1-2 inches stroke. The valve is of the balanced piston type, and is operated by a Shepherd governor attached to the fly-wheel. Quick action and regulation within 1 1-2 percent from full load to no load is secured with this governor. The engine is remarkably silent in its working and is of a type especially desirable for modern yachts and similar service.



The generator is of the eight-pole type, designed for a continuous output of 5 Kw. at 650 r. p. m. It is capable of carrying for short periods of time, 50 percent overload, without shifting of brushes or flashing at commutator, and is capable of continuous operation at 25 percent overload without sparking or undue heating. After a continuous run of ten hours, at full load, the increase in temperature above that of the surrounding atmosphere will not exceed 40 degrees C. in any of its parts. The insulation resistance is such that it will withstand a break-down test of 1,500 volts for a period of one minute and shall show a resistance of at least one megohm with an initial voltage of 500. The total weight of the set, which includes the engine, generator, and sub-base, ready for operation, is 1,140 pounds.

The Monitor Electrical Speed Recorder.

This little instrument is the outcome of the experiments of two retired engineers of the United States Navy. A simple, yet very ingeniously contrived circuit closer and breaker, is placed



on the end of the shaft, or in the shaft alley, and from this are led the wires to the instrument, which may be placed in any part of the ship. Instantly upon the engines turning over, the hand upon the dial starts marking each revolution until one minute is up, when the hand will stop, registering the exact number of revolutions per minute. It will stay at this point, unless there is a difference in the succeeding minute, in which case the hand will fall back, or advance, and accurately show the varying speeds. On the face of the instrument is an ordinary counter reading up to 10,000 revolutions, so that readings can be taken at any time. In the instrument is a time chart or disc, which revolves once in 24 hours, the face of which is divided into circles and arcs corresponding with the dial face; a swing pen is attached by positive movement, to work in conjunction with the hand upon the dial, and this dial being divided into six four-hour watches, can be taken off, dated, and filed away for future reference, thereby keeping a perfect log.

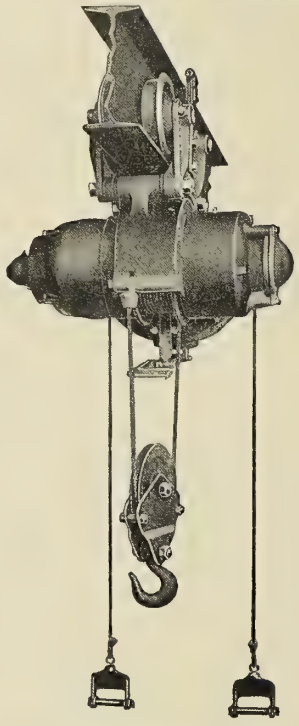
In connection with this is an automatic circuit closer and breaker, which may be placed on the taffrail, and by towing the log, or rotator, the indicator in the pilot house will at a glance show just how fast the vessel is traveling through the water, the range in nautical miles being from 0 to 38 knots; and also keep an accurate record of the same on the time chart, before mentioned. This combination, a dial of which is graduated into knots and revolutions, makes a very convenient instrument for merchant steamers, and is finding a ready sale where it has been introduced; for the master can see at a glance just how fast the vessel is traveling through the water, and by turning a switch can see also just how many revolutions the engines are making. Another unique device in connection with this is the engine annunciator, showing the direction that the engines are running. This is placed near the bell pull in the pilot house or on the rail.

Again, upon using the standard anemometer cups adopted by the weather bureau, and a specially contained circuit breaker and wiring to the same machine with a different dial, the wind velocity can be read at a glance, from 0 to 150 miles per hour or over, and an accurate record left upon the time chart, as in the other cases. It is also finding great favor with leading yacht clubs, being placed on the roof of the club house, where the cup revolutions open and close the circuit breaker, and the instrument itself, placed in the parlor of the club house, may be referred to at any time.

The instrument is an ornament to the chart room of a private yacht or house boat, is inexpensive, and can be installed at short notice by an ordinary mechanic or ship's electrician. The after care is very small, simply winding the clock once in eight days, filling the pen once a week, and putting on the chart or paper disc once in 24 hours. Masters or owners of vessels or private yachts, by corresponding with the Monitor Electrical Speed Recorder Company, P. O. Box 71, Cambridge, Mass., will be supplied with full details and particulars.

A New Electric Hoist.

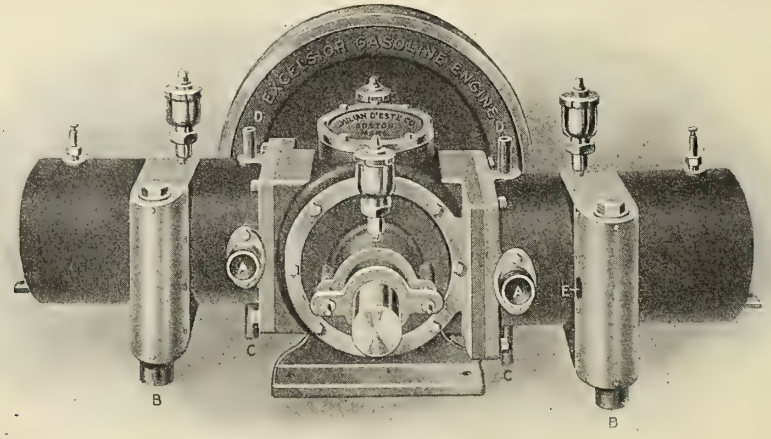
The Shepard electric hoist is constructed on lines entirely different from those of any others. It solves the problem of reducing the hoisting apparatus of the modern electric traveling crane to a self-contained and easily accessible machine, small and simple enough to be able to compete successfully in such intermittent or minor operations as serving machine tools, lifting



copes, setting cores, etc. It combines simplicity with high efficiency, strength and durability with comparative lightness. It has a speed control which will satisfy the most exacting, and the head-room required is so small that it is perfectly adapted for location under galleries, beneath belts, or in low rooms where any less compact hoist could not be used at all. The price is low enough to permit its employment with profit in any class of hoisting to which power may be applied to advantage. It is manufactured by the General Pneumatic Tool Company, Montour Falls, N. Y.

The Excelsior Gasoline Engine.

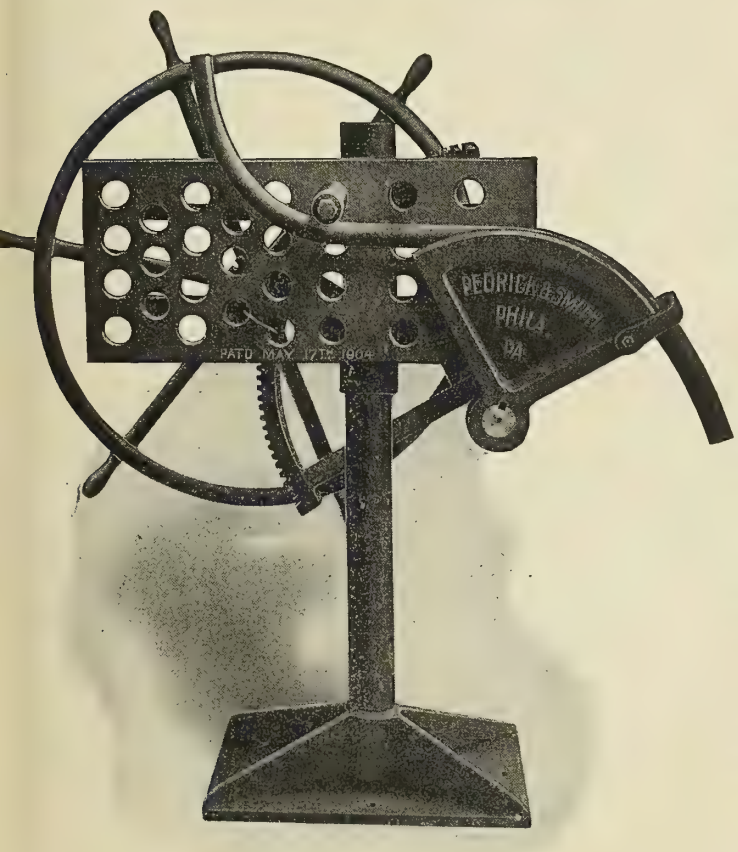
This engine is built in two distinct types: the horizontal opposed and the vertical, and is a two-cycle engine for marine work, embodying all the good features of the best marine engines, such as spark-plug ignition, no valves coming in contact with the fire to give trouble, receiving charge through induction port, car-



buretor instead of vaporizer; in addition these engines have several features of their own, such as automatic reversing device, adjustable bearings made of bronze, great speed range, arrangements to prevent back firing in crank-case, high compression (giving great efficiency), semi-automatic grease lubrication throughout, and the use of ground joints instead of packing. The greatest care and attention has been given to the correct proportioning of all parts for strength and selection of material for the greatest wear. This has enabled the builders to produce a perfectly reliable engine, durable, economical, light, compact, and powerful.

To reverse this engine all that is necessary is to throw over the lever, as in a steam engine, and the engine will reverse itself automatically. This is a very important feature, when one person is alone in handling a boat, as it gives time for attention to other things. Since the auxiliary engine is becoming almost a necessity in a sailboat which is used for business or pleasure purposes, the remarkably small amount of fuel used, and the lightness of the engines, with their adaptability for this particular work, together with the automatic reversing device, would recommend the Excelsior gasoline engine to anyone.

These engines are furnished complete with all accessories, or the engine alone, as preferred. The builders, however, recommend that the purchaser buy the engine complete, with propeller, shaft, stern bearing stuffing-box, and muffler, as the parts are made specially for the engine and the best possible results can be obtained from them. For prices and further particulars address the Julian D'Este Company, 24 Canal street, Boston, Mass.



A New Pipe-Bending Machine.

This machine, while being of a somewhat simple character, will accomplish a great variety of work in line of its intended duty. It requires the services of an ordinary helper only, who can, with the aid of this machine, bend pipe to any desired complex curvature in a very short time. The machine is operated by a hand-wheel which carries a pinion. The latter engaging a quadrant-gear operates the bending quadrant. One end of the pipe which is to be bent is held in position by a U-shaped clip while a pin or roller in the platen engages the other end. The placing of the pin or roller in the different holes governs the curvature obtained.

Being light in weight it is readily carried from shop to job, or *vice-versa,* and can be secured to any column, stanchion or any available support in a few minutes, or a suitable stand can be furnished, as shown. Piping of steel, iron, brass, copper, or other materials up to 2 inches diameter, can be bent cold. It has a great value in shipyards, pipe-shops, locomotive-works, sugar-houses, and other places where pipe is used to any extent. Makers of heating plants, structural and architectural iron workers, fire-escape manufacturers, will find this portable pipe bender especially useful. It is also well-adapted for the use of special dies that can be readily attached for bending light angles, flats or tee bars to any desired radius as easily as bending pipe. One man can, without assistance, bend a piece of 2-inch pipe to an S-bend in three minutes. The cost of repairs, where it has been used ten hours per day in shipyards, railroad shops, and other places, has been so slight in nature as to be a negligible quantity. The machine is constructed by Pedrick & Smith, of Philadelphia.

TECHNICAL PUBLICATIONS.

The Naval Constructor. By George Simpson, M.I.N.A. 588 pages, 4 by 6 inches; 288 illustrations. 1904, Kegan Paul, Trench Trübner & Company, Limited, London; and D. Van Nostrand Company, New York. Bound in leather. Price $5.00 net.

This handbook has been prepared with the object of supplying a ready reference for those engaged in the design, construction, and maintenance of ships. It contains a large amount of new and original matter, prominent in which are a chapter on design and tables of standardized fittings and details. It is divided into seven sections, of which the first treats of ship calculations, being devoted to the displacement, buoyancy, stability, launching, strength, speed, and power of ships, and the preparation of specifications. The second section deals with the strength of materials used in the construction of ships, including timber, and the various qualities of structural sections. The third part is devoted to fittings and details; the fourth to the rigging and ropes of the vessel; the fifth gives dimensions of the various minor equipments, such as boats, anchors, etc., besides giving requirements of the Board of Trade, Supervising Inspectors, and yacht-racing rules. The last two sections are devoted to tables of materials and to various mathematical tables. The work as a whole should find a ready adaptation to the needs of the up-to-date ship constructor and designer.

Dimensions of Pipe Fittings and Valves. By W. D. Browning. 82 pages. 1904, The Draftsman, Cleveland O. Price, bound in flexible cloth covers, 50 cents; in leatherette, 75 cents, postpaid.

This book contains a large assortment of useful diagrams and tables, with descriptions of the styles and uses of valves and pipe fittings. It has also useful information on the subject of steam heating and pumps. There are over fifty tables of dimensions, most of them taken from actual measurements, others compiled from trade catalogues.

Maxwell's Theory and Wireless Telegraphy. Part I.—Maxwell's Theory and Hertzian Oscillations. By H. Poincaré. Translated by Frederick K. Vreeland. Part II.—The Principles of Wireless Telegraphy. By Frederick K. Vreeland. Pages 255; illustrations 145. 1904, New York: McGraw Publishing Company. Price $2.00.

The object of this book is to give a physical treatment of Maxwell's theory and its applications to some modern electrical problems; to set forth the fundamental principles which, according to Maxwell and his followers, underlie all electrical phenomena; to show how these principles explain the ordinary facts of electricity and optics; and to derive from them a practical understanding of the essentials of wireless telegraphy. The purpose not being to

either establish or defend a theory, mathematics and abstruse reasoning have been avoided. It has been the intention to pick out the fundamentals which have stood the test of time and are now generally accepted, and to put them in such form that the busiest man may use them, or the student may take them as stepping stones to the more advanced theory.

The second part of the book has been taken up at the point where M. Poincaré dropped it, the line of thought being carried into the practical field of wireless telegraphy, and the principles laid down in Part I being applied to the various problems involved. Certain typical systems have been described, and explanations offered to show why some have failed and others succeeded, and to explain their mode of operation in the light of Maxwell's ideas. The book is not intended as a treatise on the subject, and no attempt is made to describe the myriad forms of apparatus. The object is to deal with principles and to trace the essential features of the development of the art.

Suction Gas. By Oswald H. Haenssgen. 88 pages; 4 illustrations. 1904, The Gas Engine Publishing Company, Cincinnati, O.

The development of a system of prime movers operated by suction or producer gas has attained much greater advancement in Europe than in the United States, where large supplies of liquid or gaseous fuel are at hand for the use of internal combustion engines. Recently, however, the subject has acquired a good deal of interest in the United States, and individuals have endeavored to investigate the general subject and to obtain reliable information on the economy to be obtained from the use of producer gas. It is believed that this work is the first reliable book upon this subject, and that it ought to fill a considerable place where information is required regarding the design, operation, cost of running, development, and possibilities of this type of gas producer.

Cyclopedia of Applied Electricity. Published in five volumes containing about 2,200 pages, over 3,000 illustrations. Bound in three-quarters red leather. List price $30.00; special introductory price $18.00. Published by the American School of Correspondence, at the Armour Institute of Technology, Chicago, 1905.

This work is one of the most comprehensive works on electricity and its practical applications that has yet been published, and should prove invaluable to electricians, engineers, telephone and telegraph operators, engineering students, and any person who has occasion to use electricity in any form. To be able to turn quickly to any phase of electricity as used in our great industries is of immense value to both the busy practical man and the engineering student. In these volumes the reader finds a treatment of the latest developments of electricity by acknowledged authorities, fully indexed, and written in a simple, direct way, free from difficult mathematics, and especially adapted to the needs of the busy practical man.

The work is not addressed to the technically educated specialist, and for this reason all of the descriptions and explanations which it contains have been simplified to the utmost possible degree consistent with a thorough presentation of the subjects treated. Higher mathematics have been entirely omitted, and the subject matter of the work is explained in simple language in connection with a large number of diagrams prepared especially for the text. The half-tone engravings are of special value as they illustrate all of the electrical machines and apparatus made by the prominent electrical companies in the United States, and therefore acquaint the reader with the types of apparatus used in current engineering practice.

The subject matter is divided into chapters or sections with a view to using the work as a text book, and the closely related subjects are brought together under a number of general headings. A great deal of attention has been given to practical examples, and each volume is supplemented with a list of review questions by means of which the reader can test for himself the knowledge which he has acquired of the subjects treated.

The usefulness of the cyclopedia as a reference book is augmented by the addition of a subject index contained in the fifth volume. The practical value of the work as a whole can hardly be questioned, as the cyclopedia embodies the various electrical courses which have been successfully used by the American School of Correspondence in teaching thousands of electricians. The work is, in fact, the outcome of a most successful method for the education of the busy man. It will be found of the greatest value to designers, constructors, and operators of electrical machinery, and as it is the only text book and reference work covering every department of electricity thoroughly, its use will save many valuable hours of search among scattered text books on electricity.

QUERIES AND ANSWERS.

All reasonable questions concerning marine engineering received from and signed by subscribers will be answered by the editor in this column. All communications must bear the name and address of the writer.

Q. 275.—I wish you would please give me some information in your next issue on the following subject:

A light draft boat is under construction here; they claim it will draw 10 inches light; the boat is 50 feet long, 15 feet beam and has two 30-inch tunnels aft, 12 feet long, open at bottom, and a 28-inch screw in each tunnel. They claim the screw in turning forces the air out, and the tunnels naturally fill up with water, and they get as much power as if the screw were in deep water. How is this, is it right? H. T. Y.

A.—The claim that screw tunnels on a light-draft steamer will fill up with water when the boat is in rapid motion is quite correct. The stream lines and the action of the propeller combined operate to raise the level of the water in the tunnel to a point considerably higher than that outside, and unless the top of the tunnel is very much above the general surface of the surrounding water the tunnel will completely fill, and the propeller will be working in an unbroken mass of water.

Q. 276.—Please answer in your next issue the following questions:

Show how an indicator diagram will be affected by a faulty setting of either eccentrics or valves, or leakage past the high-pressure slide valve; supposed to be taken from a triple-expansion engine. P. L. S.

A.—(1). When the angular advance of the eccentric is too small all the operations of the valve are too late. The admission lines slope inward at the top, compression corners are too small, and release so late that the steam cannot reach exhaust pressure until the next stroke is nearly completed. This state of affairs is illustrated in Fig. 1.

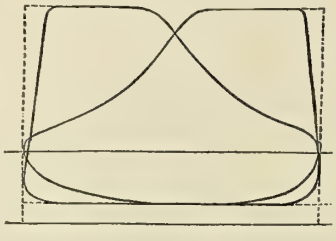


FIG. 1.

(2). When the angular advance of the eccentric is too great the operations of the valve are all too early, with resultant enormously exaggerated compression curves.

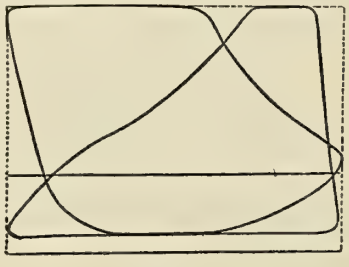


FIG. 2.

(3). When the eccentric is properly set, but the valve is incorrectly placed upon the rod, the lap at one end of the stroke will be excessive and at the other end deficient. The valve will therefore at one end admit steam too soon and continue the admission too long, giving too late a cut-off, and resulting in a diagram similar to that at the left side of Fig. 2. Release is here too late, and the exhaust period shortened. The right-hand diagram shows the effects at the other end to be quite the reverse: steam admission late and too short, cut-off too soon, exhaust commencing early and continuing late.

(4). Leakage past the valve results in the continued admission of steam after the cut-off has occurred, thus reducing the effective expansion and increasing the mean effective pressure. In some cases a part of this steam leaks into the exhaust side and raises the back pressure. Both results are detrimental to economy. When this leakage occurs in the high-pressure cylinder of a multiple-expansion engine, the leakage in itself is not so serious a matter as with a simple engine, as the steam which does no work in the first cylinder may still be used in a later one; but the effect on economy is much the same sort of thing.

Q. 277.—Would you please answer the following: What would be the best arrangement of air, feed, and circulating pump for a steam launch of 26 I. H. P.? Would it be best to have each driven independent, or the three by one steam cylinder? What size air pump would be required for the above horsepower? Also circulating pump? Or perhaps you can tell me of some practical book which has this in it; also the price. L. W.

A.—For small boats, such as the case to which you refer, the favorite arrangement is to include the air and circulating pumps in one combination and to operate the feed pump independently. If you consult the catalogues of leading pump makers, you will find listed various arrangements of combined air and circulating pumps, mounted either with the condenser or separately. You should have a circulating pump capable of handling about 75 gallons of water per minute at a reasonable speed, with a surface condenser of 50 to 60 square feet of surface, and an air pump with a displacement capacity of about 10 cubic feet per minute.

Your feed pump should be capable of handling at moderate speed 6 to 7 gallons of water per minute.

The approximate size of such pump would be about as follows:

Combined air and circulating pump; steam cylinder 5 1-2 inches, air cylinder 6 inches, circulating pump cylinder 6 inches, stroke 7 inches.

Feed pump: steam cylinder 3 1-2 inches, water cylinder 2 1-4 inches, stroke 4 inches.

These estimates are based on your stated horsepower, and assume that you have a fairly economical simple condensing engine. If your engine is compound you will use somewhat less feedwater and will require correspondingly less condensing water, but it would not be wise to cut very much the dimensions suggested.

Q. 280.—Would you kindly answer a question for me through your Q. & A. columns?

I would like to know if there are any ferryboats on the East River, New York, that can go ahead with one paddle-wheel and back with the other simultaneously? If so, kindly give name of same, and by what company they are owned, and at what piers they land. J. D.

A.—There are no such boats in operation.

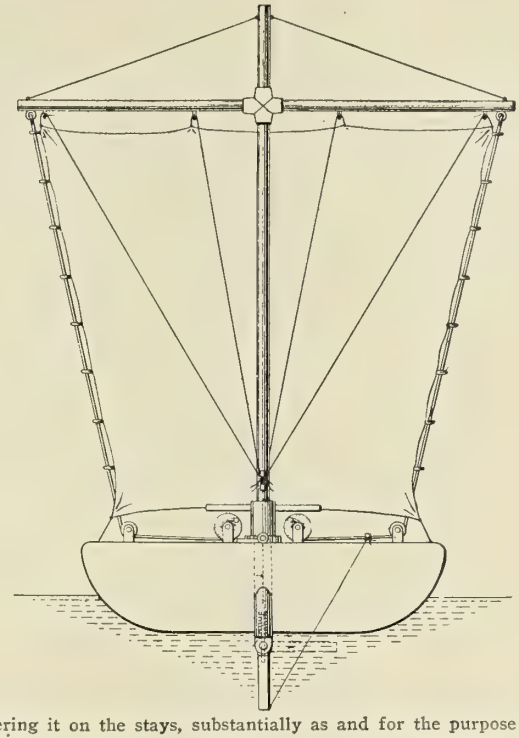
SELECTED MARINE PATENTS.

780,027. DREDGING APPARATUS. JOSEPH EDWARDS, BROOKLYN, N. Y., AND WALTER H. GAHAGAN, BOONTON, N. J.

Abstract.—The invention therefore consists, essentially, in a conical or cylindrical hood or envelop surrounding the mouth of the suction pipe and firmly connected thereto in such a manner as to provide an annular space between the suction pipe and the outer hood, said hood extending upward alongside of the suction pipe from the mouth as far as may be necessary to prevent that caving in of the materials around the mouth as will be likely to smother the suction and clog it from receiving the proper amount of water used in the process of hydraulic dredging. Two claims.

779,440. SAIL BOAT. JAMES P. POOL, BROOKLYN, N. Y.

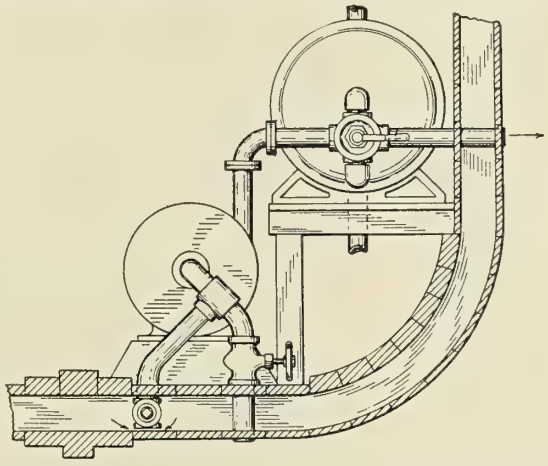
Claim.—2. A boat provided with a suitable mast, a square or wing-and-wing sail, an adjustable stay arranged at each of the side edges of said sail for trimming the sail, the said sail being provided with tackle for raising



and lowering it on the stays, substantially as and for the purpose set forth. Three claims.

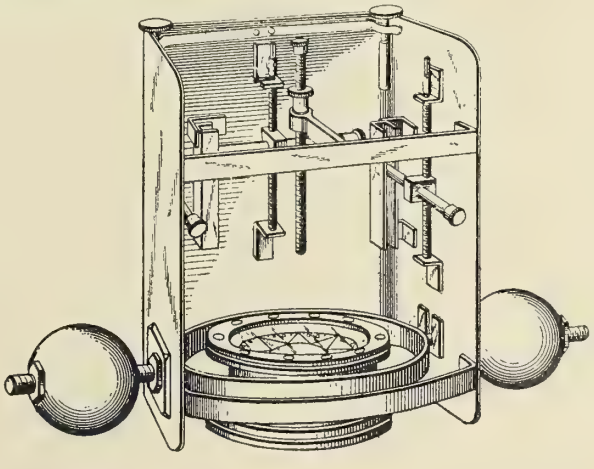
779,889. REVERSIBLE-CURRENT APPARATUS FOR CONDENSERS. JOHN TODD, SEATTLE, WASH., ASSIGNOR OF ONE-HALF TO GEORGE W. WILSON, SEATTLE, WASH.

Abstract.—This invention relates to marine condensers and circulating pumps, and more particularly to apparatus whereby the current of water passing through the condenser may be quickly reversed in order to dislodge



obstructions collected in the mouths of the condenser tubes and convey such accumulations into the outboard water-discharge pipe, and also to provide means whereby in an emergency the bilge water may be pumped directly into the said discharge pipe without having to pass through the usual channels. Two claims.

780,374. BINNACLE FOR SUBMARINE BOATS. JOHN S. NEGUS AND HERBERT BLOSSOM, NEW YORK, N. Y.

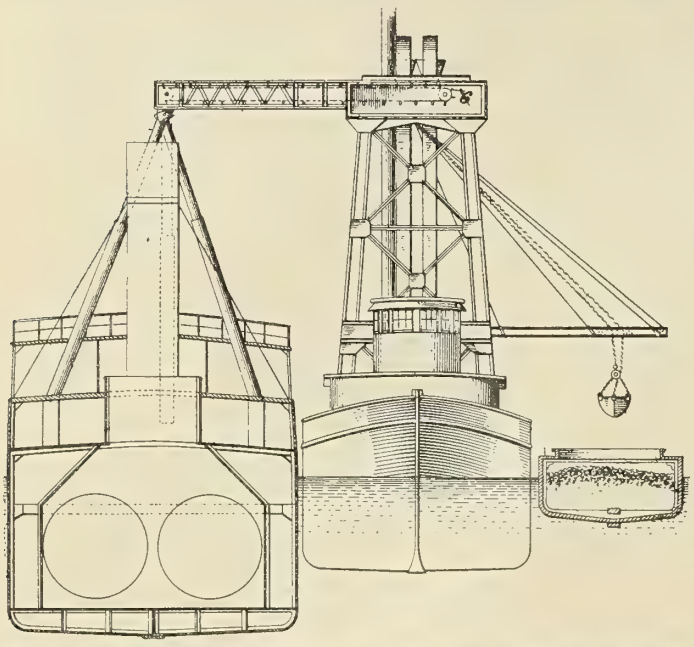


Claim.—5. A binnacle, having a compass pivotally mounted therein, horizontally disposed magnets mounted on the binnacle above the level of the compass, and means for moving said magnets toward and from the compass, said means and the magnets being mounted in a readily-removable manner and being accessible from above the binnacle.

6. A binnacle, having a casing and quadrantal correctors at respective opposite sides of same, said corrector comprising a screw-threaded rod projecting out laterally from the casing, a ball which is threaded on said rod, and means for securing said ball in place when set on the rod. Nine claims.

780,083. VESSEL FOR BUNKERING AND LOADING SHIPS. EDWARD T. WILLIAMS AND GEORGE A. ORROK, BROOKLYN, N. Y.

Claim.—1. A self-propelling, non-cargo-carrying vessel, having a supporting tower combined with a hoisting apparatus adjacent to the base of the tower for raising cargo or bunkering material lying alongside and discharging it inboard; an elevator supported by the tower and to which the discharged material is delivered and which is adapted to raise the material to a height

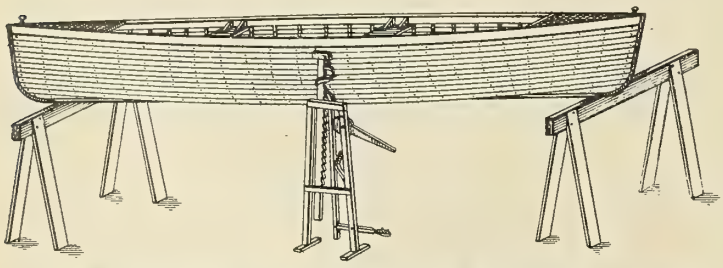


directly over said vessel less than would suffice for its descent by gravity therefrom to the farthest point that the vessel is adapted to ultimately deliver the material; an arm whose inner end rests directly upon and is secured to the upper end of the tower and whose outer end extends outward therefrom; a driven conveyer carried by said arm for taking the material delivered to it by the elevator at this height and conveying the same outboard beyond the opposite side of the said vessel to a point directly over the loading or bunkering ship; and a delivery chute leading downward from this point to the ship. Two claims.

781,052. FOLDING BOAT. LYNN S. DEAL AND ADDISON D. GUTCHES, KALAMAZOO, MICH.

The objects of the invention are, first, to produce a folding bottom or keel for boats having a flexible skin, so that the same may be folded into a compact form; second, to produce a folding boat bottom which will contain the greatest strength when unfolded consistent with lightness; third, to so construct a boat bottom that it will rest from end to end on the framework of the boat to which it is applied; fourth, to produce a folding boat bottom in which the end sections fold into the central section, so that the slats forming the end sections lie in spaces between the slats forming the central section, the slats of both sections lying in the same plane when folded. 9 claims.

781,542. BOAT-BUILDER'S RIVETING JACK. JOSEPH D. MORLEY, LAKE PLEASANT, N. Y.



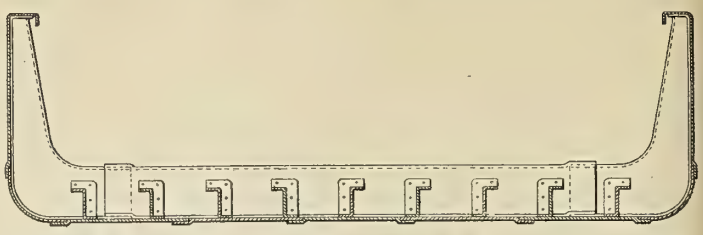
The object of the invention is to provide a jack for properly supporting in respect to the strakes of the boat the weight against which the rivets are clenched, so that the man who is driving the rivets will have both hands free. 5 claims.

782,153. BARGE OR OTHER BOAT. JAMES S. MARTIN, PITTSBURG, PA.

Claim.—1. A barge formed from pressed steel having a series of longitudinally-disposed bottom ribs, side ribs terminally attached to the outermost bottom ribs, the latter ribs having upper flanges extending over the lower ends of the side ribs and a sheathing secured to said ribs.

2. A pressed-steel barge composed of a series of bottom and side ribs, the bottom ribs being arranged in planes at right angles to the side ribs and having upper and lower angular flanges, cross-girders arranged at intervals and having enlarged terminals embracing portions of and secured to the side ribs, said girders intersecting the longitudinal bottom ribs and having the flanges of the latter secured thereto, and sheathing plates connected to the ribs.

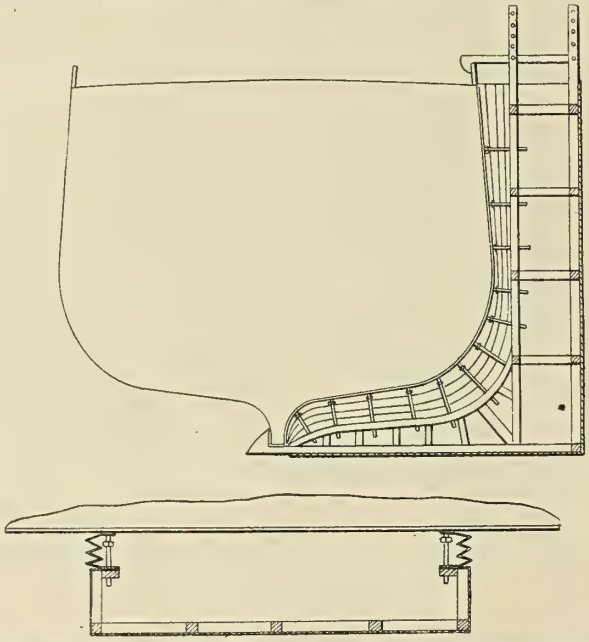
3. A pressed-steel barge composed of a series of bottom and side ribs, having end flanges, the bottom ribs being disposed longitudinally, and an outer sheathing connected to said ribs and having the upper terminal thereof bent



at an angle in relation to the upper extremities of the ribs to provide a supporting means. 3 claims.

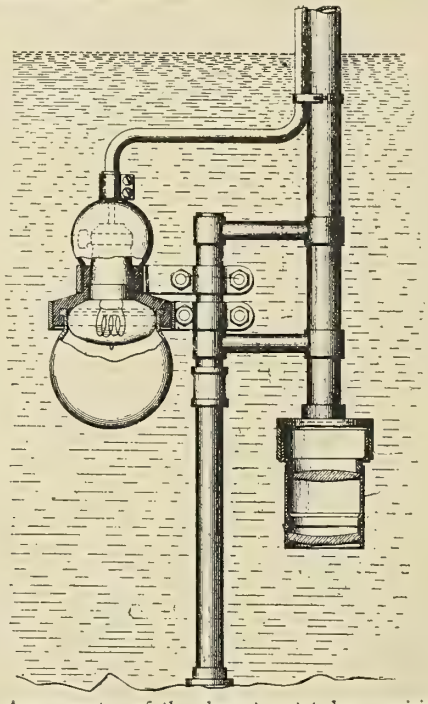
783,276. SHIP REPAIR MECHANISM. JOHN C. HUGHES, SOUTH OMAHA, NEB.

This invention has particular relation to means for permitting access to the sides and bottom of boats, ships, or vessels of any type without the neces-



sity of removing them from the water, and consists in a caisson or removable closure capable of being attached to the sides and bottom of a vessel and means for forming a water-tight joint between the casing and the vessel, and thus making it possible to remove the water from about that portion of the vessel, so that repairs may be made. 12 claims.

783,730. SUBMARINE VIEWING APPARATUS. FRANCIS MCMAHON, SOMERVILLE, MASS.



Claim.—1. An apparatus of the character stated, comprising a sight-tube provided with a lens immovable relatively to the upper portion of the tube, an electric lamp adjacent to the lens and at one side of and connected with the tube, said tube and lamp being adapted to be inserted in a body of water, and means for limiting downward movement of the apparatus, said means being located at one side of the line of sight through the tube and lens whereby a direct view downward through the tube and lens will not be obstructed by said limiting means. 3 claims.